The Association
of
Engineering and Shipbuilding
Draughtsmen.

HIGH PRESSURE HYDRAULIC CONTROL EQUIPMENT.

By J. RODGER and R. HARTKOPF, M.I.E.S.

Published by The Draughtsman Publishing Co., Ltd.,
96 St. George's Square, London, S.W.1.

SESSION 1945-46.

Printed by Milne, Tannahill & Methven, Ltd. (T.U.), 12-14 Mill Street, Perth.

ADVICE TO INTENDING AUTHORS OF A.E.S.D. PRINTED PAMPHLETS.

Pamphlets submitted to the National Technical Sub-Committee for consideration with a view to publication in this series should not exceed 9,000 to 10,000 words and about 20 illustrations, making a pamphlet of about 40 to 48 pages. The aim should be the presentation of the subject clearly and concisely, avoiding digressions and redundance. Manuscripts are to be written in the third person.

Drawings for illustrations should be done either on a good plain white paper or tracing cloth, deep black Indian ink being used. For ordinary purposes they should be made about one-and-a-half times the intended finished size, and it should be arranged that wherever possible these shall not be greater than a single full page of the pamphlet, as folded pages are objectionable, although, upon occasion, unavoidable. Where drawings are made larger, involving a greater reduction, the lines should be made slightly heavier and the printing rather larger than normal, as the greater reduction tends to make the lines appear faint and the printing excessively small in the reproduction. In the case of charts or curves set out on squared paper, either all the squares should be inked in, or the chart or curve should be retraced and the requisite squares inked in. Figures should be as self-evident as possible. Data should be presented in graphical form. Extensive tabular matter, if unavoidable, should be made into appendices.

Authors of pamphlets are requested to adhere to the standard symbols of the British Standards Institution, where lists of such standard symbols have been issued, as in the case of the electrical and other industries, and also to the *British Standard Engineering Symbols and Abbreviations*, published by the B.S.I. in 1934 at 3/6. Attention might also be given to mathematical notation, where alternative methods exist, to ensure the minimum trouble in setting up by the printer.

The value of the pamphlet will be enhanced by stating where further information on the subject can be obtained. This should be given in the form of footnotes or a bibliography, including the name and initials of the author, title, publisher, and year of publication. When periodicals are referred to, volume and page also should be given. References should be checked carefully.

Manuscripts, in the first instance, should be submitted to the Editor, *The Draughtsman*, 95 St. George's Square, London, S.W.1.

For Pamphlets, a grant of £15 will be made to the Author, but special consideration will be given in the case of much larger pamphlets which may involve more than the usual amount of preparation.

The Association
of
Engineering and Shipbuilding
Draughtsmen.

HIGH PRESSURE HYDRAULIC CONTROL EQUIPMENT.

By J. RODGER and R. HARTKOPF, M.I.E.S.

Published by The Draughtsman Publishing Co., Ltd.,
96 St. George's Square, London, S.W.1.

SESSION 1945-46.

Printed by Milne, Tannahill & Methven, Ltd. (T.U.), 12-14 Mill Street, Perth.

List of Photographs, Diagrams and Sketches.

INDEX

HIGH PRESSURE HYDRAULIC CONTROL EQUIPMENT.

By J. Rodger and R. Hartkopf, M.I.E.S.

INTRODUCTION.

The use of liquids under pressure for transferring power from a central source to a number of machines has—from the earliest days of engineering—occupied a position of considerable importance. In spite of the great advances which have taken place it is still supreme in many applications in the shipbuilding and heavier industries, and is assuming great importance in a new industry, namely plastics. The smooth supply of power, together with flexibility and ease of control, make it ideal for heavy presses and similar machinery, where the weight of the reciprocating parts and the intermittent nature of the movement required result in considerable stresses and considerable wear if any mechanical form of drive is employed.

The satisfactory working of hydraulic machinery depends to a very great extent on the provision of a suitable method of controlling the supply of power, and it is probably in this direction that the greatest advances have taken place. It is the hope of the writers that the following drawings and descriptions of the more important aspects of control equipment in general use will be of particular assistance to those who are concerned with its uses and potentialities, and that they will provide a basis for the development of any special purpose gear should the necessity for it arise. The worked examples included are intended to give some illustration of the methods of approach to various problems, and the appendix contains information which is often difficult of access at short notice.

The writers wish to express their appreciation of the great kindness shown to them by Messrs. E. Bruce Ball, M.A., M.I.Mech.E., M.I.E.S., of Glenfield and Kennedy ; L. A. Ruff, M.I.E.S., of Hugh Smith & Co. (Possil) Ltd., and F. H. Towler, M.I.Mech.E., of Towler Brothers (Patents) Ltd., who have generously provided information and photographs and have given every assistance. In addition, special thanks are due to Mr. Alex. C. Livingston, A.R.T.C., for his assistance in checking the manuscript and the examples, and providing valuable suggestions and information.

In conclusion, the writers wish it clearly to be understood that, except where acknowledgment has been made, the sketches and calculations are purely typical and do not, for obvious reasons, include any refinements and alterations which may be made by individual manufacturers in any type of valve.

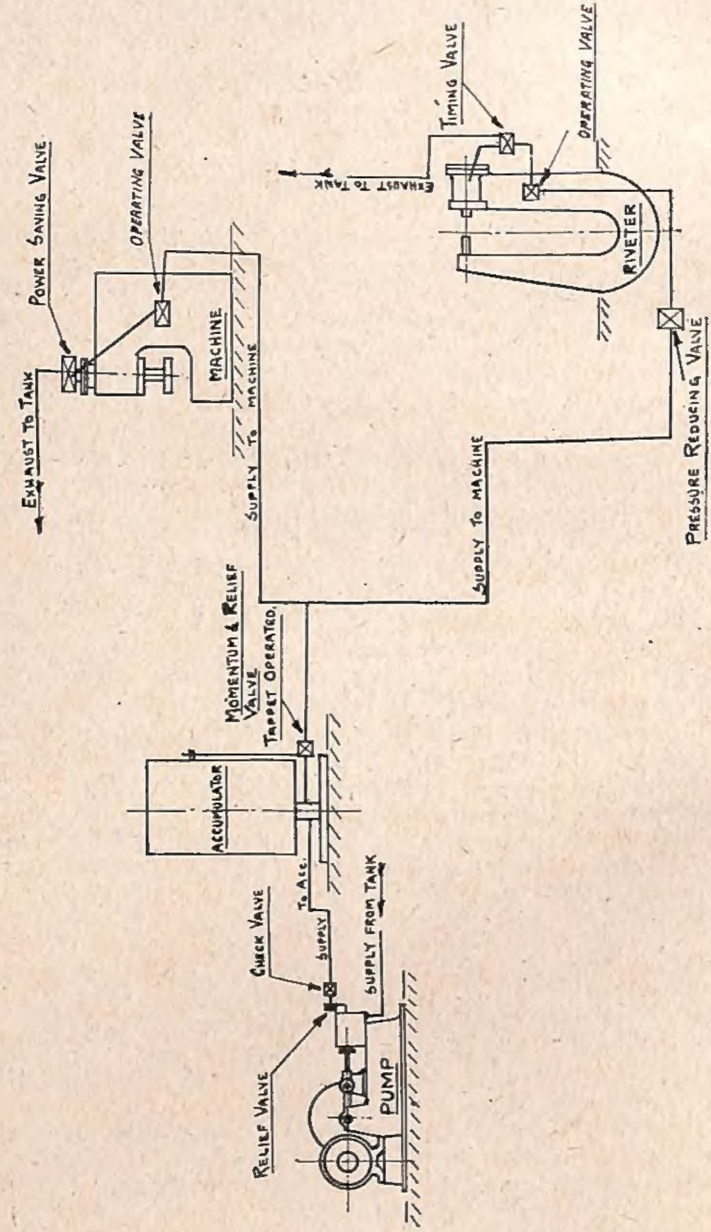


Fig. 1.—Diagrammatic Layout of Hydraulic System.

Section 1.

THE HYDRAULIC SYSTEM.

Before discussing control gear in detail, it is necessary to examine it with reference to the nature of the plant and the type of work it is required to perform. In this connection it is of interest to note that of recent years hydraulic engineering has developed in two distinct directions; and for those who are not familiar with hydraulic practice a short description of these trends is given below.

The older and still the more common use of hydraulic power is to supply a number of machines from one central source of high pressure with a capacity sufficient for all the machines in the workshop or yard. A diagrammatic layout of a simple example of this kind of installation is shown in Fig. 1.

In this system the hydraulic power is usually provided by one or more sets of electrically-driven three-throw pumps which have a capacity varying from about five gallons per minute up to about five hundred gallons per minute on the largest installations. It is of interest to note that, as this type of pump and the associated system of piping has an almost indefinite working life, it is advisable when considering an installation to allow for any extra pumping capacity which may be required in the future and to design the system accordingly.

The pressure at which the water enters the piping from the pump varies from about 800 lbs./sq. in. upwards, but there has been of late a steady tendency towards higher pressures of 1500 lbs. and 1 ton per sq. inch. As the power which a given quantity of fluid can transmit is directly proportional to its pressure, it will readily be seen that a higher pressure will allow of a considerable saving in the size and weight of the equipment as a whole. In some special cases still higher pressures are used, and Fig. 2 gives some idea of what is commercially practicable.

But whatever the pressure which is delivered by the pumps, they are almost always of the constant delivery type and it is therefore necessary to provide some means of relating the constant delivery of the pumps to the intermittent demands of the various machines. This function is performed jointly by the accumulator, relief valve and the tappet switches mounted on the accumulator. When the demand from the machines falls below the amount supplied by the pumps the excess fluid flows into the accumulator and raises the weighted casing. If this continues for a sufficient time a kicker on the casing throws over a switch and stops the electric motor driving the pumps. As the stored fluid is used, the accumulator casing gradually falls until a second kicker operates another switch. This causes the electric motor to start again and the procedure is repeated. The relief valve is usually employed as a precaution against the failure of the electric gear, and, when

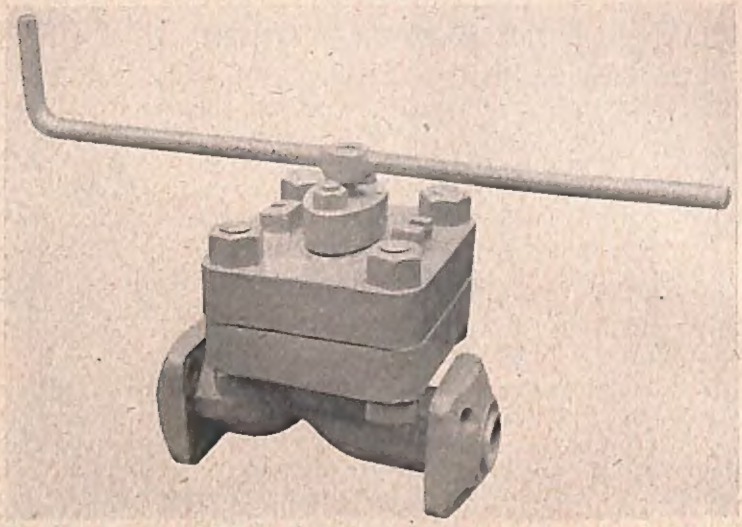


Fig. 2 shows a typical 3-in. diameter cast-steel hydraulic screwdown stop valve with special square flanges suitable for a working pressure of 2½ tons per sq. inch and a test pressure of 4 tons per sq. inch. This valve is fitted internally with gunmetal working parts, forged bronze spindle, and cover bolts of high tensile steel. Operation is by means of a long key. It is of interest to note that the unbalanced load on the closed disc of this valve amounts to some 16 tons, and to relieve this heavy load a pilot valve is incorporated in the main valve as described above, arranged so that the pressure is equalised at the first turn of the spindle.

Glenfield & Kennedy.

operated it by-passes the supply from the pumps and returns it to the exhaust system. If, however, it is not desired to stop and start the pumps in this manner the relief, or unloading valve, as it is sometimes called, can be used without the tappet switches.

The foregoing short explanation serves to show how a sufficient and steady supply to the machines can be assured, and once this is achieved, their actual working requirements are controlled by the gear attached to the machine itself. As this will be discussed in detail in the following sections, it is not proposed to deal with it here, but to conclude with a description of the second trend of development in hydraulic engineering.

This second trend of development which, although of recent origin is becoming increasingly important, is to dispense with the central pumping and distributing system and to mount an individual pump on or beside each machine. It must be realised, of course, that these pumps are vastly different from the large three-throw constant delivery pumps mentioned above. They work at far higher speeds (1500 R.P.M. is quite common) and are often smaller

in size than the electric motor which drives them. The hydraulic medium used in these pumps is always oil, a light mineral oil having a viscosity of about 100 sec. redwood at 70°F. being usual, and the greatest care must be taken to keep air, water and solid impurities from entering the system. Another interesting feature is the absence of any type of glands or packing, experience having shown that under these conditions the working clearances can be made fine enough (about ·0003″ on diameter) entirely to prevent leakages even at pressures as high as 5000 lbs./sq. inch.

At this point it should be stressed that any comparison between the two systems can only lead to wrong conclusions and a mistaken idea of their respective applications. The system using a central high pressure supply is designed for use in heavy engineering industries shipbuilding being a typical example. From the one source of supply and using the cheapest and most easily replaceable hydraulic medium there is, namely water, comes the power for presses, bending, straightening and joggling machines, riveters, hoists and frame setting machines, and innumerable other types of machines, some of them light and portable and others capable of doing the heaviest work there is. In spite of the fact that many of these machines are in open yards exposed to all weather conditions they will work year in and year out with only the slightest attention. They will stand ill-treatment and abuse which would ruin any other type of machine and are yet extremely simple to operate, and as long as this type of heavy work exists this type of machine will be required to perform it.

The single machine system, on the other hand, has been developed along with the growth of one of the most modern industries, namely the die moulding of plastic and similar substances. The aim here is to provide high-speed machines suitable to mass production methods, and which, by virtue of their use of hydraulic power, have more flexibility and can be controlled with greater precision as regards pressure, timing, etc., than would be possible with any mechanical system.

The development of this recent trend in hydraulic engineering has enabled hydraulic methods to play a very useful part in the production and moulding of plastic materials. A further application of this type of system is shown in Fig. 3.

Although the limitations of space make it impossible to do more than touch on the more general applications of hydraulics, the question of hydraulic remote control gear is at least of sufficient importance to merit some mention, and for the sake of those interested, a short description of it will be given in a later section.

With the above points in mind the various types of high-pressure hydraulic control gear will be discussed in the following sections and examined in relation to their suitability for particular functions in one or other of the systems.

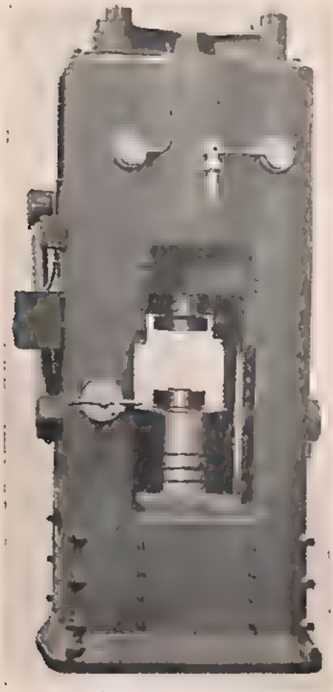


Fig. 3 shows a press for forging automobile valves at the rate of 300 per hour. Some of the hydraulic rams in this machine move at a speed of 20 ft./sec., and the forging stroke is completed in less than 1/100 sec. This disproves the old idea that hydraulic presses are necessarily slow.

Towler Brothers (Patents) Ltd.

Section 2.

THE ROTARY FACE OR DISC TYPE VALVE.

The rotary face, or disc type valve, is one of the most useful types of general purpose control valve to be found in the centrally-supplied hydraulic system which uses water as a hydraulic medium. It is the logical development of the older type of hydraulic slide valve, which in turn was derived from the well-known steam engine slide valve. A sectional illustration of the rotary face type valve is shown in Fig. 4.

It will be seen from the figure that it consists of a metal block known as the body, and in this are ports through which the water enters and leaves the valve. Where these ports enter the valve, recesses are turned and studs provided for bolting on the flanges fixed to the piping. In small sizes, however, a screwed union can

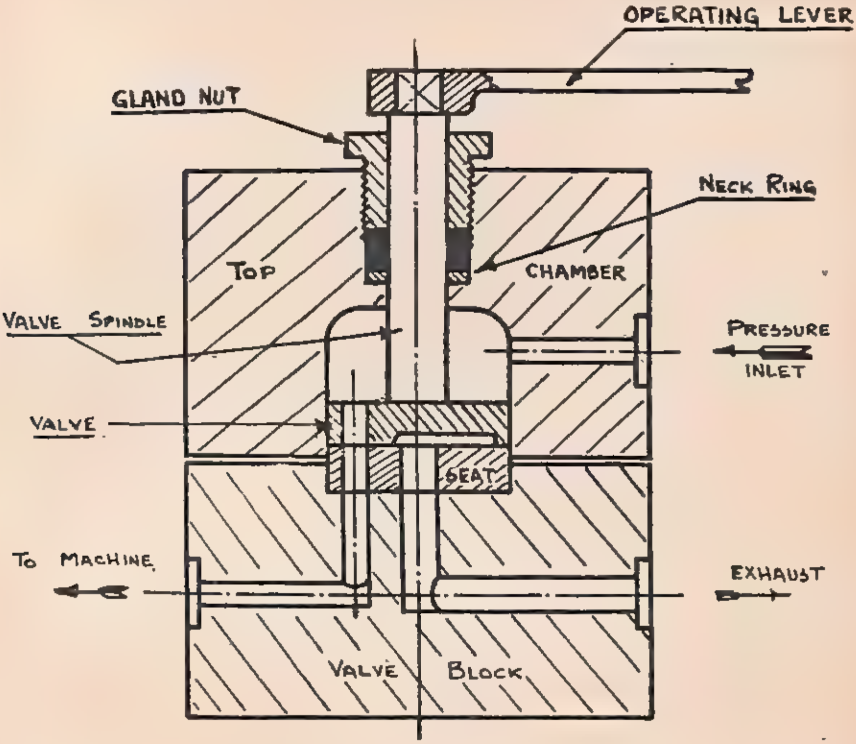


Fig. 4.—Sectional Illustration of Rotary Face or Disc Valve. Single Acting.

be provided instead. The body is machined to accommodate the valve seat, and the top half of the body contains the necessary gland and packing for the operating spindle. This spindle is attached to the operating lever and provides the means whereby the valve can be rotated on the seat. As the spindle is turning in the packing it is mechanically unsatisfactory to make spindle and valve in one piece. If, however, some form of flat is machined on the spindle and fitted into a corresponding slot in the valve, the water pressure will hold the valve evenly on the seat, as described below. In this case a collar would have to be fitted to the spindle to prevent it from being forced out by the water pressure. The action of the valve is very simple ; the rotating valve is held hard against the seat by the action of the high pressure water which is permanently connected to the chamber above ; and as it turns, the various ports in the valve and seat coincide, allowing the water to pass through to the appropriate port. In the position shown, the high pressure is connected to the "to machine" port, and it will

be seen that a rotation of 180° would bring the valve into the "exhaust" position.

When the operating lever is turned to "exhaust" the water from the machine is returned to the valve and directed through the exhaust port back to the supply tank in the system. This exhaust connection is made by means of a slotted section on the underside of the valve, and the force with which the high pressure water above holds the valve against the seat prevents this high pressure water from leaking between the valve and seat directly to exhaust. From this it can be seen that any wear which takes place cannot affect the efficiency of the rotary face valve, as the high pressure water will continue to keep the rotating valve firmly seated.

It might be mentioned here that the ram of the machine can be returned to the top of its stroke by the provision of a small area of constant high pressure on the underside of the ram. Fig. 10 in the worked example at the end of this section illustrates this.

Once the principle of the rotary face valve is understood it is easy to see that by a suitable arrangement of the ports a large number of different purpose valves can be made, and the fact that it is not necessary to alter the sizes and shapes of the spindle, valve block, etc., is, naturally, a considerable advantage. Figures 5, 6, 7, 8 show the modifications in the valve and seat required for some of the types of rotary face valves which are in general use.

The valve and seat in Figure 6 are for the double-acting type. As explained above, it is usual in a small machine to have the ram returned to the top of its stroke by means of constant pressure applied to the underside. If, however, positive control in both directions is required, the valve can be arranged as shown, and pressure can thus be applied to either side at will, while the opposite side of the ram is automatically connected to exhaust.

In certain cases it is desired to operate the ram from an individual pump, and some method must be provided for by-passing the water when the machine is not being operated, preferably by unloading the pump. This can be done by the use of the valve and seat shown in Fig. 7, which is similar to the double-acting arrangement but which has an extra port, which, when the operating lever is in the "by-pass" position, by-passes the pressure water direct to exhaust.

A self-explanatory arrangement for a simple selector valve is shown in Fig. 8, and although space does not permit of more examples, it is felt that those given will enable the readers to see the possibilities of other types to meet special requirements.

The rotary face valve is particularly suited to small type portable machines, for if the port connection to the machine is brought out through the bottom of the block the complete valve can be bolted on to the machine, making a strong, neat and efficient

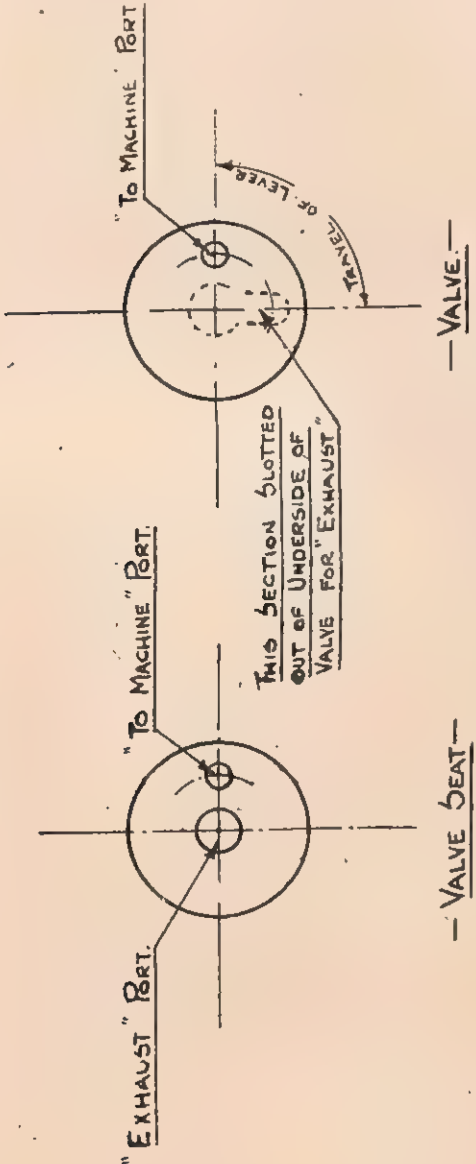


Fig. 5.—Valve and Seat for Single-Acting, Rotary Face Valve.

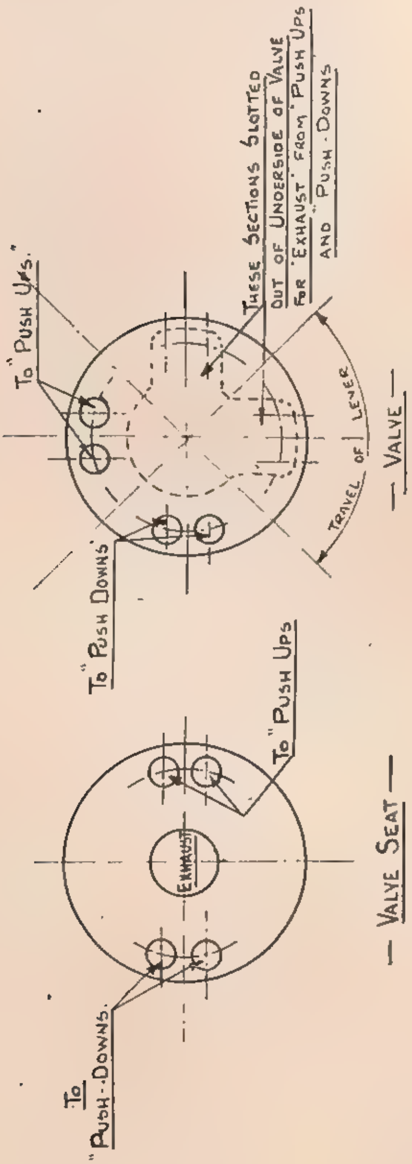


Fig. 6.—Valve and Seat for Double-Acting Rotary Face Valve.

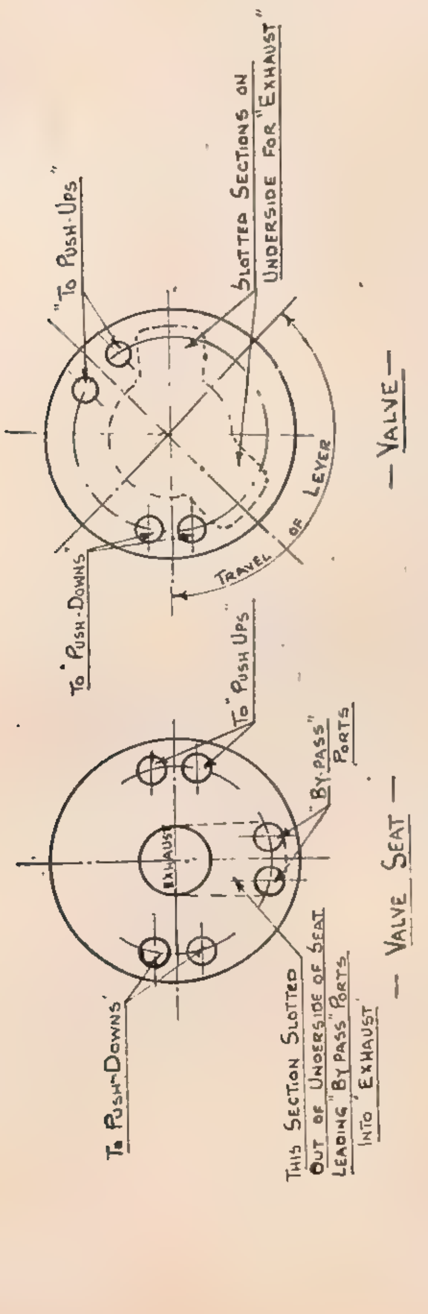


Fig. 7.—Valve and Seat for Double-Acting Rotary Face Valve with By-Pass.

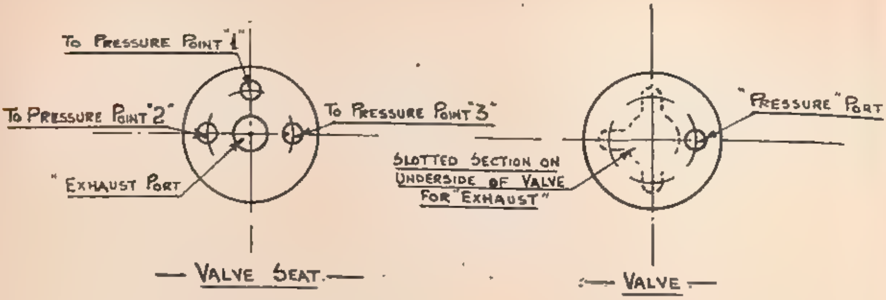


Fig. 8.—Valve and Seat for Typical Rotary Face Selector Valve.

connection, without the necessity for piping. Fig. 9 shows a rotary face valve mounted in this fashion on a portable frame setting machine.

The limitations to the use of the rotary face valve lie in the fact that the area of the ports through which the water must pass are relatively small, and if a large amount of water has to be used its speed through the valve and the resulting head losses may become excessive. This means that if a water speed of about thirty feet

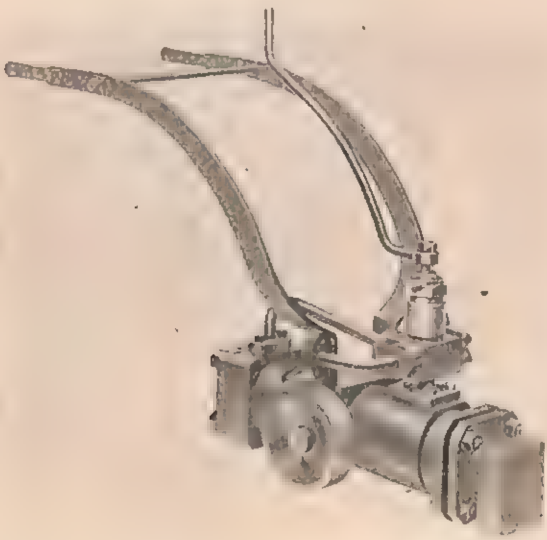


Fig. 9 shows a Rotary Face Valve mounted on a Portable Frame Setting Machine. The valve has been mounted on the machine, making a compact and sturdy arrangement. The constant pressure supply pipe to the far end of the ram is rigidly attached, and the operating lever has been extended for ease of operation.

Hugh Smith & Co. (Possil) Ltd.

per second is adhered to, the speed of travel of the ram of the machine must decrease in proportion to the increase in area of the ram. It is, of course, possible to make a larger valve, but here again there are definite limitations. As the diameter of the ports in the valve and valve seat are increased, the area and the pressure on the valve seat increase very rapidly (roughly proportionally to the square of the increase on the diameter) and the valve rapidly reaches a point where the increased friction renders manual operation of the valve impossible. For similar reasons the valve tends to lose some of its usefulness at the higher pressures over 3000 lbs./sq. inch.

This disadvantage can be partially compensated by the use of the semi-balanced type of rotary face valve. By anchoring the spindle to the valve, the pressure on the seat is reduced by an amount equal to the force which the high pressure water in the top chamber exerts in tending to push the spindle upwards. Although, by varying the diameter of the spindle any desired pressure between the valve and seat can be obtained, it must be remembered that the pressure cannot be reduced beyond certain definite limits because of the danger mentioned above, that the high pressure water will leak across the face of the valve. The absence of lubrication accentuates the friction, and although lubricated face valves have been produced and are quite successful, they require a certain amount of attention which, in practice, unfortunately, is not often given ; and the non-lubricated valves—particularly in the smaller sizes—are very widely used, and give excellent results.

In actual practice the diameters of the valves and seats used vary from about 1½″ to 3″, the smaller sizes being by far the most common.

Within the limitations mentioned, the rotary face valve is an extremely useful type. It has only one or two moving parts, is very simple to operate and gives very accurate control over the amount of water passing. In addition, although a clean oil as a hydraulic medium immensely increases the life of any valve, rotary face valves have been known to work, using water from the mains as a hydraulic medium, for periods up to twenty years without even replacing the valve and seat. In fact, when some of these valves have failed at long last the cause of the failure has been found to be, not as one would expect, due to the valve and seat having worn away, but to the fact that the water, after many years, had actually corroded its way through the body just underneath the valve seat and was running to exhaust, while the valve and seat themselves remained in quite workable condition although, of course, very considerably worn.

The following worked examples cover the most important aspects concerning the design of this type of valve. A by-pass valve has been selected because it introduces some problems not

met with in the other types, but the calculations can be applied equally well to any variation of this type of valve.

Example.—A rotary face valve is required for a 15-ton general purpose press, having a ram speed of 3 ft. per minute, supplied with water at 2240 lbs./sq. inch, from an individual constant delivery pump.

Allowing for packing friction, a ram to the sizes shown in Fig. 10 would be quite suitable.

(*Note.*—The R.F. valve is frequently used for operating far larger machines, but this is done in conjunction with special automatic control gear and will be discussed in a later section).

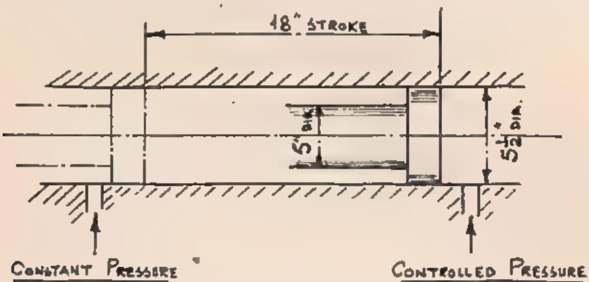


Fig. 10.—Diagram of Ram Referred to in Calculations.

If the stroke of the press is 18″, we have :—

Capacity required for full working stroke =

Area of cylinder × stroke

$= 23{\cdot}75 \times 18$

$= 427{\cdot}5$ cubic ins.

As the speed of the ram is 3-ft. per minute, the working stroke of the machine will occupy 30 seconds, thus :—

Quantity of water per minute required

$427{\cdot}5 \times 2 = 855$ cubic ins./min.

This quantity must be passed through the port of the valve and seat. Assuming a ¼″ diam. port, we have an area of 0·049, say 0·05 sq. ins.

Therefore, speed of water through port

$855/0{\cdot}05 = 17{,}100$ inches/minute.

$= 1425$ feet/minute.

$= 24$ feet/second approximately.

This would be quite satisfactory as the maximum recommended speed of water through the port is 30 feet/sec.

When the pump is being by-passed through the valve under no load conditions, the same quantity of water will clearly have to be passed, and it would be reasonable to assume that this size

of port would be suitable here also. It is common practice, however, to make the by-pass port somewhat larger, as it is desirable to have low speeds in the exhaust system, since head losses here are much more troublesome than the same head losses in the high pressure side.

Assuming a $\frac{3}{8}''$ diam. by-pass port, we have an area of 0·11 in., therefore the speed through the by-pass port

$$\begin{aligned} 855/0{\cdot}11 &= 7772 \text{ ins./minute.} \\ &= 647 \text{ ft./minute.} \\ &= 10 \text{ ft./sec. approximately.} \end{aligned}$$

As can readily be seen, a $1\frac{1}{2}''$ diameter valve and seat will be sufficient to accommodate these ports, *i.e.*,

One $\frac{1}{4}''$ diameter port "to machine."
One $\frac{3}{8}''$ diameter port "by-pass."
One slotted section for leading the returning water from the machine to exhaust.

Here it is worth noting that it is usual to give the exhaust port a slight lap or lead on the "to machine" port in all double-acting valves, as this removes the possibility of getting pressure on the one side of the ram before the other side has been connected to exhaust.

The final consideration is that of the force required at the operating lever to operate the valve.

Assuming first the valve is not balanced.

The total load on the valve face = area of valve × W.P.

Area of $1\frac{1}{2}''$ diam. × 2240 lbs./sq. in.
1·76 × 2240 = 3943 lbs. approx.

Taking the coefficient of friction factor, which will, of course, vary with the materials used, as 0·1 (see note below) we have :

Force to be overcome = 3943 × 0·1 = 394 lbs. approx.
Assuming a lever arm of 12" radius, then
Force required to operate = 394/12 = 33 lbs. approx.

If, however, the valve is to be semi-balanced, using a 1" diam. valve spindle, then the load on the valve face

(Area of valve, less area of spindle) × W.P.
1·76 sq. ins. – 0·78 sq. in. × 2240 lbs./sq. in.
0·98 × 2240 = 2195 lbs.

Therefore force to be overcome = 2195 × 0·1 = 220 lbs. approx.
Using a 12" radius lever as before, then
Force required to operate = 220/12 = 19 lbs. approx.

The above method of calculating the operating force on the lever appears at first sight to be open to some criticism. The friction of the packing around the valve spindle has been neglected, and the calculation of the total load on the valve face has been based on the assumption that the whole of the area of the underside

of the valve face is at zero pressure. The explanation lies in the fact that this is an empirical method based on experience, and the coefficient of friction factor used is not necessarily the actual coefficient of friction for the materials involved, but a factor determined by experiment which allows for the points mentioned above.

This method is quick and simple to use, and in practical applications gives a perfectly satisfactory degree of accuracy.

Section 3.

THE MUSHROOM TYPE VALVE.

There can be few engineers who are not familiar with one form or another of the mushroom or mitre type valve, particularly in connection with internal combustion engines. This type of valve, however, works just as well with liquids as it does with gases, and plays a very important part in hydraulic control work. In its most common application it performs the function of checking any reversal in the direction of flow of the liquid in the system, and valves of this type are installed in centrally supplied systems as a safety device, particularly where a number of accumulators are used. The valve is placed in the line between the supply pumps and the accumulators and thus if any of the connections at the pumps should fail, or the valves accidentally jam, the water stored in the accumulators would be prevented from returning into the pump. If, as is often the case, the pumps are automatically stopped when the accumulator reaches the top of its stroke, the check valve affords additional protection against the possibility of leakage during the time the pumps are not working.

As it is often convenient to have a stop valve at the same point in the system, it is the usual practice to combine the two in the form of a valve known as the screwdown stop check valve. It should be noted that it would be very dangerous to have any form of stop valve in such a position that it can block the supply from the pumps. A relief valve should always be placed between the pump and any stop valve to bypass the water if the stop valve is inadvertently closed when the pumps are running. Fig. 11 shows a typical example. The action is quite simple and very effective. When the spindle is in the "up" position, the valve is free to lift and close in the same way as an ordinary check valve. By screwing down the spindle, however, the valve is held hard down against the seat and the valve then becomes an ordinary stop valve.

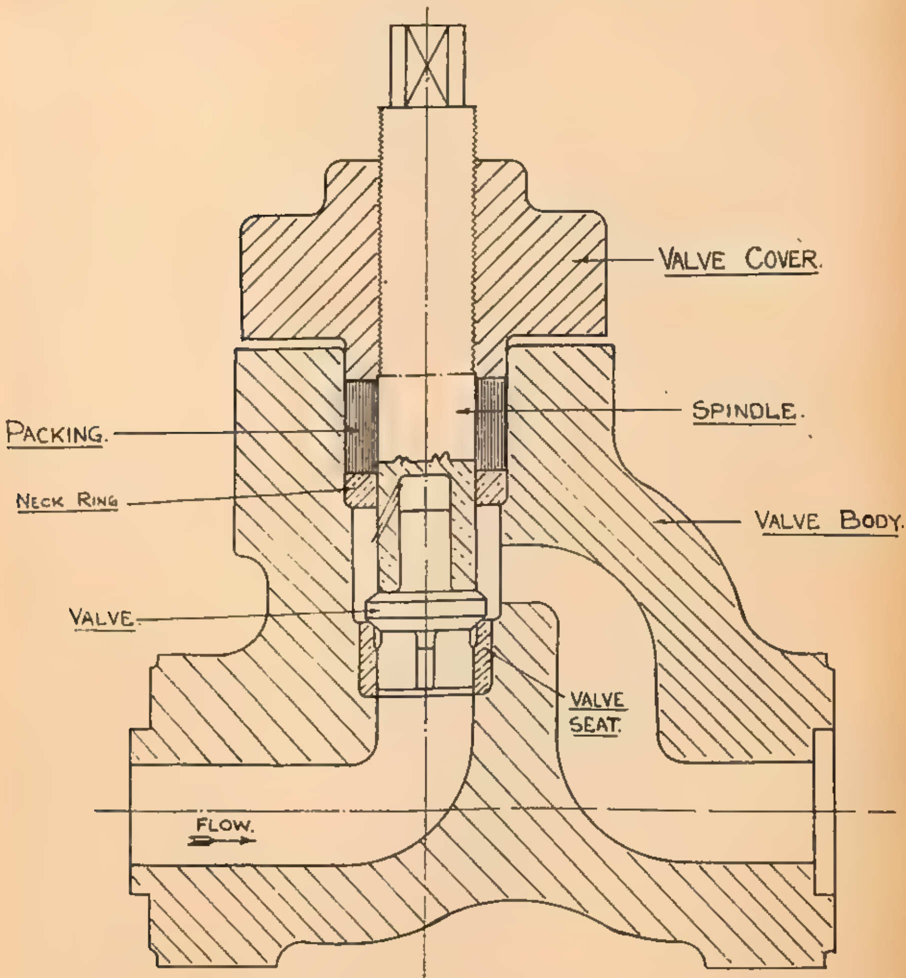


Fig. 11.—Stop and Check Valve.

While the check valve will stop any reversal of flow from taking place, it cannot prevent an excess pressure being built up due to a block in the system or some similar reason, and thus it is necessary to provide some means of regulating the maximum pressure so that it never rises above that which the system can safely stand. This is done by means of a relief valve, and a valve of this type is universally used in connection with all types of hydraulic systems. One such valve can almost always be found mounted directly on the pressure side of the supply pump. Fig. 12 shows a typical relief valve of the weight loaded type, which is used mostly on accumu-

lators. The valve is of the simple "straight through" type and the two exhaust openings shown are merely to enable the valve to be fitted in the exhaust line, thus saving piping and tee pieces. The diameter of these ports must, of course, be large enough to prevent any restriction of the flow in the exhaust line. Where, however, a relief valve is required, not in connection with an accumulator, it is more often of the spring-loaded type. The spring is mounted directly over the spindle and takes the place of the weight in keeping the valve in its normal shut-off position.

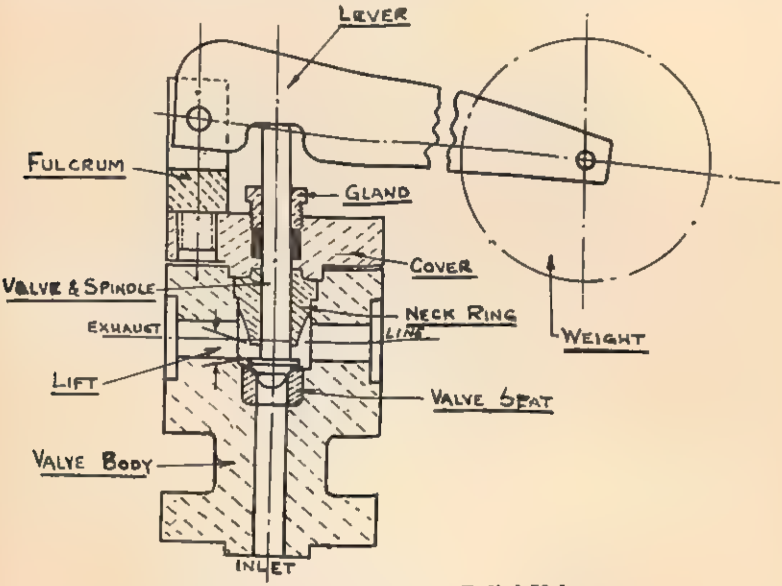


Fig. 12.—Weight Loaded Relief Valve.

The weight-loaded type in Fig. 12 serves a double purpose when used with an accumulator, as it not only prevents an excessive rise in pressure from taking place, but by fixing a vertical rod to the lever of the valve and providing suitable kickers and tappets, the accumulator casing can be made to lift the lever and thus open the valve when the accumulator reaches the top of its stroke, thus giving a positive protection against over-run should the switches normally arranged to cut off the power from the motor driving the pump, fail to function.

From the figure it will be clear that the weight or spring must be sufficient to hold the valve shut at all pressures up to the maximum allowable in the system. Immediately the pressure becomes greater than this, the resultant extra load on the underside of the valve will lift it from its seat and allow water to exhaust to the

supply tank until the pressure drops to normal and the valve closes again.

There is another condition which often arises in a hydraulic system, particularly where several machines are being fed from long pipe lines. The water travelling along the pipe is suddenly stopped when an operating valve is closed, and it is clear that this, or a similar condition in the system, would be capable of setting up a surge pressure which could be large enough to burst the cylinder of one of the other machines or otherwise damage the system. In one actual case the surge pressure was found to be three times the normal working pressure. To overcome this, a fitting capable of absorbing this energy is used. It is known as an alleviator or momentum valve, and because a relief valve at this point is often necessary, the two valves are frequently combined in the form called a momentum and relief valve.

Fig. 13 shows a section of a typical momentum and relief valve, and as can be seen it is very similar to an ordinary spring-loaded

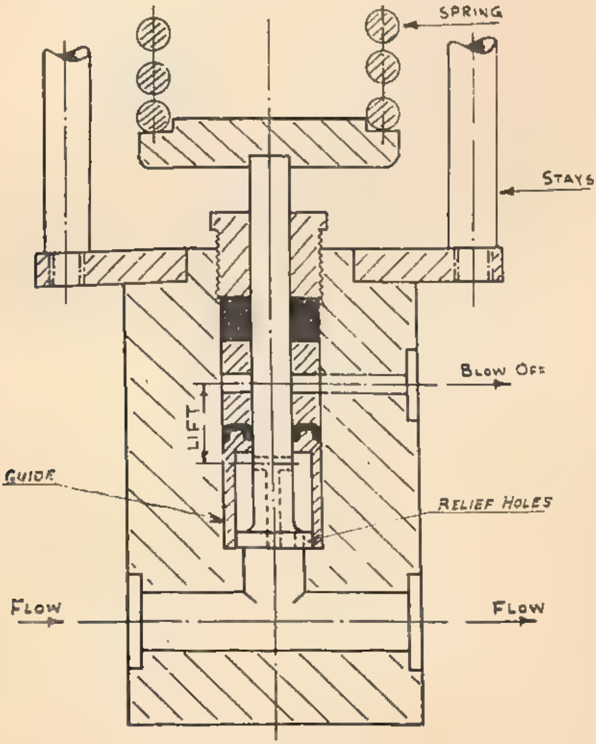


Fig. 13.—Typical Momentum and Relief Valve.

relief valve, with the difference that the spindle has to travel a considerable distance before the relieving part of the valve is able to act. When a sudden surge takes place the energy is absorbed in lifting the valve against the reaction of the spring.

The relatively slow return of the valve as the pressure returns to normal distributes the energy over a period of time and thus prevents any sharp rise in pressure from taking place. The travel of the valve is made large enough to allow this to happen without it reaching the blow-off point, but should the pressure remain above normal the valve will then act as an ordinary relief valve. The alleviator is merely a similar device without any provision for blowing off, but otherwise its action is exactly the same.

The following are typical calculations intended to give an idea of the general principles of design in connection with the valves explained above. It will be noticed that considerations such as packing friction, materials, etc., are not mentioned, partly for the sake of simplicity and partly because these are matters of special knowledge and experience on the part of individual makers rather than questions of general design principles. In order to fix the size of a relief valve, the general procedure is as follows :—

$$Q = Cd\, A \sqrt{2\, g\, H}$$

is a well-known hydraulic formula giving the quantity of water discharged from an orifice, where

Q is the quantity of water passed in cubic feet/sec.
Cd is the coefficient of discharge (in this case about ·24).
A is the area of the orifice in square feet.
H is the head in feet.

Converting this into a more convenient form, we have

$$A = \cdot 0315\, Q/Cd \sqrt{P},$$

where

A is the area of the orifice in square inches.
Q is the quantity of water passing in gallons per minute.
P is the pressure in lbs./sq. inch, and
Cd is the coefficient of discharge as before.

Then supposing it is desired to fix the size of a relief valve to be fitted in a system passing 20 gallons per minute at a pressure of 1500 lb./sq. in.

$$\begin{aligned} A &= \cdot 0315\ Q/Cd \sqrt{P} \\ &= \cdot 0315 \times 20 / \cdot 24 \times \sqrt{1500} \\ &= \cdot 068 \text{ square inches.} \end{aligned}$$

This is the area which is required if the valve is to pass the full amount of water in an emergency. In order to minimise any hunting action, it is usual to make the lift of a relief valve not more than 1/20 of the diameter.

This, if d = the required diameter of the valve, then

$$
\begin{aligned}
\text{Area of valve} &= \pi d \times \text{lift (approx.)} \\
&= \pi d \times d/20 \\
&= \pi d^2/20 \\
\therefore \qquad d^2 &= 20\text{A}/\pi \quad \text{and } d = 2{\cdot}5\sqrt{\text{A}} \text{ (approx.).} \\
\text{Thus} \qquad d &= 2{\cdot}5 \times \sqrt{{\cdot}068} \\
&= {\cdot}65'', \text{ say } \tfrac{3}{4}'' \text{ dia.}
\end{aligned}
$$

Thus a standard $\frac{3}{4}''$ relief valve would be quite suitable for the purpose.

While the mathematics dealing with surge pressure and water hammer are rather complex and outside the scope of a work of this nature, it is interesting to note that in the formula derived by Allievi the surge pressure is independent of the length of pipe (*i.e.*, the column of water). "The Theory of Water Hammer," by L. Allievi, is recommended to those who wish to study this aspect of hydraulics.

The design of an alleviator, or momentum valve, however, is somewhat simplified, for, as the very purpose of this fitting is to prevent any considerable rise in pressure, the questions of the elasticity of the water and the piping do not arise.

In general there are two main positions in which this fitting can be used. The first and most important is the position shown in Fig. 14. where an alleviator is fitted near an accumulator. When the full supply is being drawn, the accumulator will be descending at a rate of about 1 foot/sec., and it will possess a kinetic energy due to this motion. This can be calculated from the formula,

$$\text{K. E.} = \text{WV}^2/2g \text{ lb. ft. where}$$

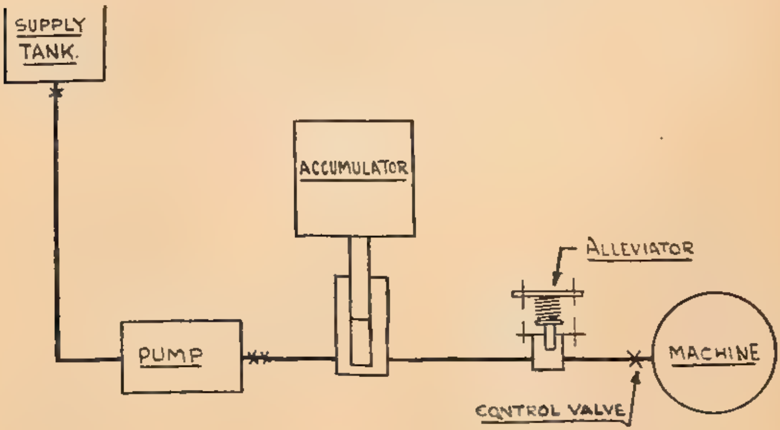


Fig. 14.—Diagrammatic Layout of Hydraulic System for Alleviator Calculations.

W is the weight of the moving part of the accumulator in lb.
V is the velocity in feet, and
g is acceleration due to gravity (32·2).

As the weight of the accumulator in larger sizes can run into hundreds of tons (see Section 5), it is clear that this kinetic energy can be considerable.

From the formula $P = W v/g t$, where

P is the uniform retarding force in lb.
W is the weight of the moving part of the accumulator in lb.
v is the velocity in feet/second.
t is the time taken for the accumulator to come to rest in secs.

We see that the pressure rise in the system is inversely proportional to the time taken for the accumulator to come to rest. The purpose of the alleviator is to lengthen this time interval, and Fig. 15 gives an idea of the pressure changes in the system with

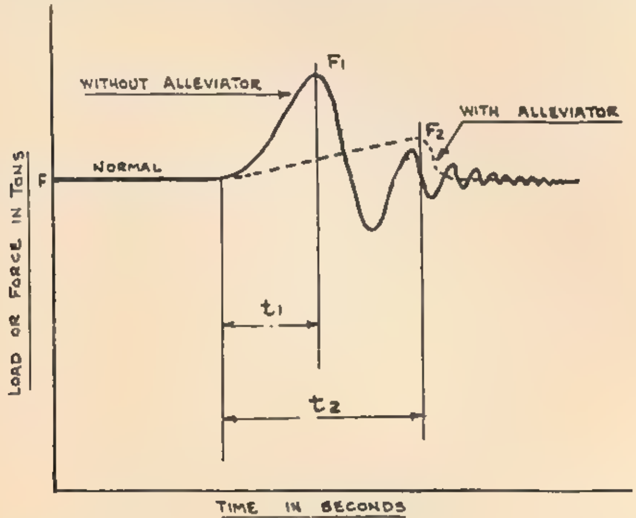


Fig. 15.—Typical Graph showing Surge in System with and without an Alleviator.

and without an alleviator. In this diagram the normal pressure before and after closing the control valve is the same, but in the case shown in Fig. 14 there would be a rise of up to 20% in the pressure, due to the fact that the accumulator would begin to rise again. (See Section 5). Another point worthy of attention is that without an alleviator the pressure variations are of an oscillatory nature, while with an alleviator the variation is steady. Readers who have a knowledge of electrical phenomena will see here an interesting parallel in the mathematics dealing with critical

resistance in an oscillatory circuit, for in both cases it is the resistance or frictional losses and the time factor which determine whether or not these oscillatory impulses occur.

It should be noted that Fig. 14 is merely diagrammatic and that the supply line from the accumulator usually feeds a number of machines. Thus the chance that they will all be shut off simultaneously is very slight, and the maximum change in the supply cannot be estimated without regard to the system concerned. In many cases empirical formulae have been derived by individual manufacturers to take the above-mentioned factors into account, and beyond stating that they are based on the considerations mentioned, it is not possible to give more detailed treatment without confusion arising between general principles and particular applications.

The second position in which an alleviator is used is at the end of a long supply line feeding one or more machines. The treatment is very similar to that given above, but in this case the kinetic energy of the water in the pipe is considered and the alleviator designed to absorb this. Here again the treatment must take into account the remainder of the system, including the surge pressure which the accumulator may set up and the possibility of all the machines being shut off simultaneously.

The "Homeyard" Type Operating Valves.

In spite of their usefulness in many hydraulic applications, there are a number of difficulties in the way of the application of mushroom or poppet type valves to the purpose of operating various machines. Even if only a single-acting valve is required, the cylinder of the machine must be alternately connected to pressure and then to exhaust. This entails the use of two valves and it is clear that if they are both opened at once the pressure water will go straight through to exhaust. This precludes the use of two separately-operated valves for normal purposes, and on the other hand if they are ganged together any wear has a tendency to cause considerable leakage. The pressure on the valve is also a factor contributing to the difficulty.

These troubles, however, have been overcome with considerable ingenuity in the patent "Homeyard" hydraulic control valve. It employs two mushroom valves in the single-acting, or single-ported, type, and four in the double-acting type, shown in Figs. 16 and 17 respectively. The action of the single-acting valve is as follows :—The valve spindles B are both shown in the shut position in the figure, and it will be noted that the bottom half of the spindle is formed into a ram. The pressure downwards on this serves to balance the upward pressure on the valve when it is closed and, as the stem of the spindle is the same diameter as this ram, it will also be balanced when open. Thus in both positions the valve

is automatically balanced at all pressures. The second major difficulty, that of ensuring that the valve is positively seated, is overcome by means of the small ram H, which takes up wear by applying a constant downward pull through a crosshead G through links F. and on to the operating lever C, by means of crosshead E. The force with which the valves are held on their seat can be accurately controlled by modifying the size of the ram H., which, of course, is permanently connected to the high pressure. When the operating lever is moved up, the pressure spindle is lifted and the other, the exhaust spindle, acts as a fulcrum. When the

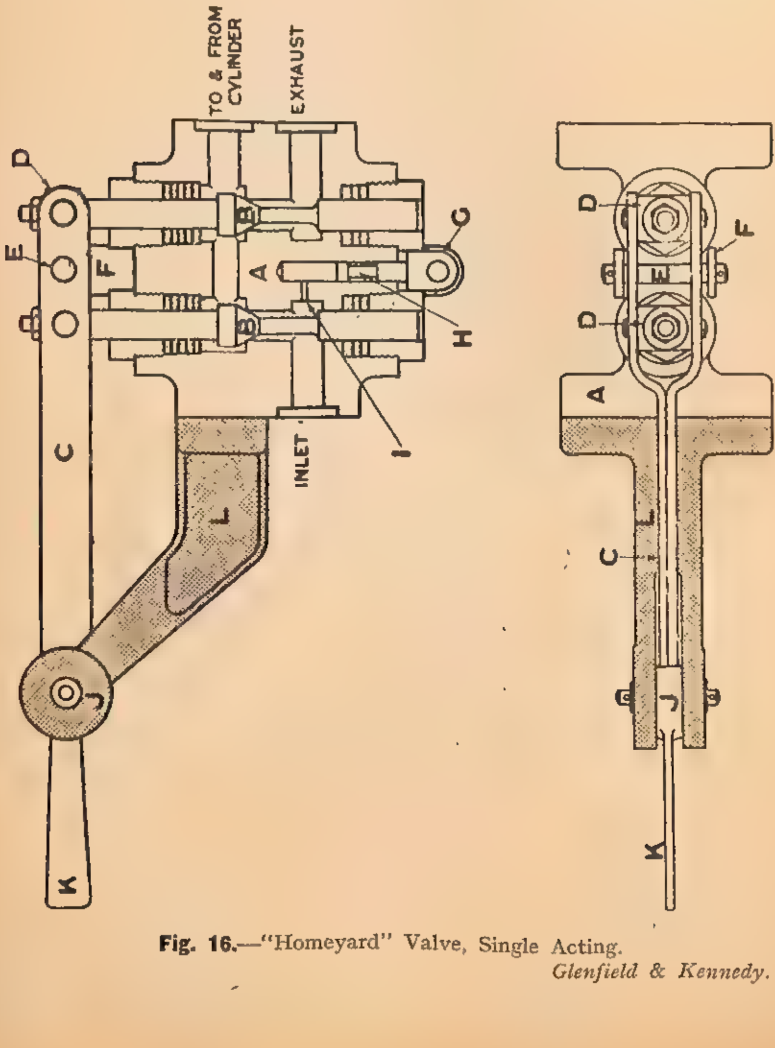


Fig. 16.—"Homeyard" Valve, Single Acting.

Glenfield & Kennedy.

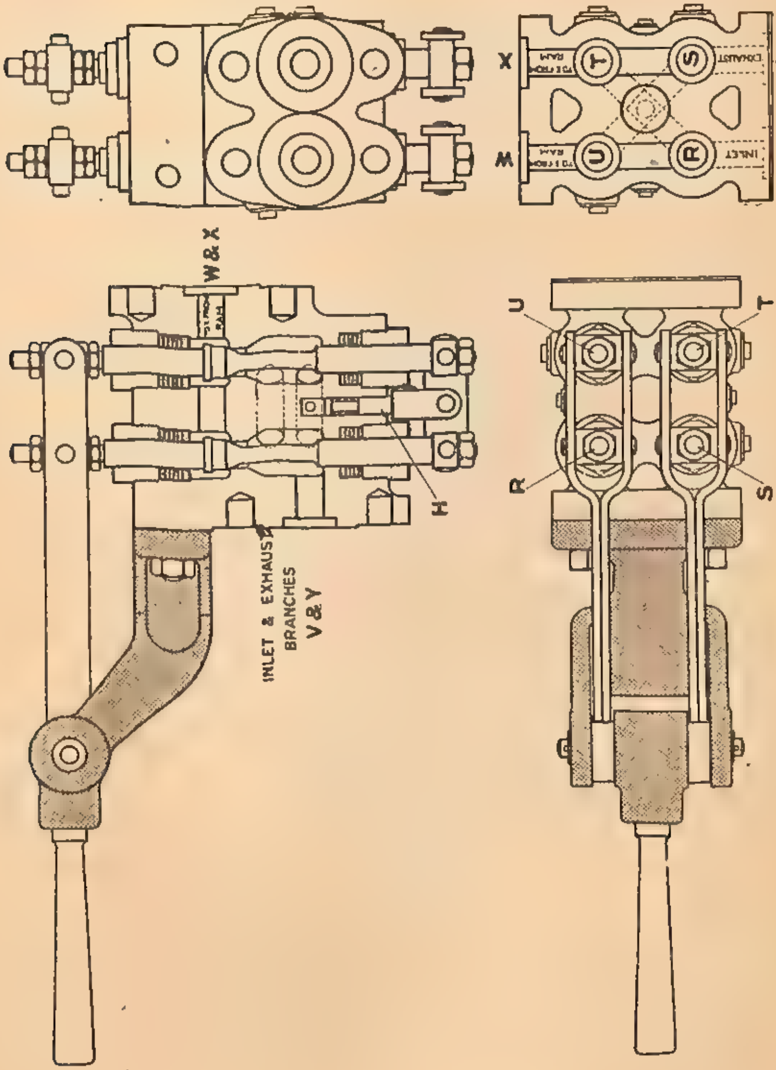


Fig. 17.—"Homeyard" Valve, Double-Acting.
Glenfield & Kennedy.

operating lever is moved the other way (*i.e.* down) the exhaust valve is lifted in a similar manner. Thus there is no possibility of the two valves being off their seats at once, and the difficulty mentioned above is overcome. In the single-acting type the inlet, exhaust and cylinder flanges are all in the same plane and also, by altering the position of the port I, the inlet and exhaust flanges can be reversed, although the connection to the machine must always be in the same position.

The plan view in Fig. 17 will assist in showing the difference between the single and double acting valve. The double-acting valve, as explained in the previous section, supplies pressure and exhaust alternately to either end of a double-acting ram. It can be seen that the four valves are cross-connected, and whichever way the lever is moved one valve is supplying pressure to the one end while the second valve is exhausting the other end ; the other two valves, of course, being held firmly on their seats. The movement of the lever in the reverse direction reverses the pressure and inlet supplies to the machine.

A final point worthy of mention is that for normal purposes the operating lever is worked by means of a handle K which has a cam plate pivoted at J. As well as ensuring the greatest ease of control, this enables the valve lever to be locked in either the up or down position as desired. In other cases, however, such as the air-operated type, shown in Fig. 18, the cam and handle arrangement is dispensed with.

The most important qualities of the "Homeyard" valve can be summarised as follows :—

The large amount of fluid which this valve can pass makes it suitable for the direct operation of quite large machines and for other purposes where a considerable flow is necessary. At the same time the small movement of the lever required fully to open the valve makes it very suitable for use with automatic or remote control. Fig. 18 gives an interesting example of this and shows a single-ported "Homeyard" valve operated by compressed air. It will be noticed that in this instance the handle is dispensed with and the valve lever is connected directly to the vertical spindle on the diaphragm. Fig. 26 is another example showing a "Homeyard" valve arranged so that the energising of the solenoid above releases a catch and allows the spring to close it. This is an instance of the use of the valve for an emergency "shut-off" arrangement and in this case the valve is opened by hand.

Although this valve is suitable for water or oil, and works well at all pressures, it is necessary to ensure that the medium used is as far as possible free from impurities and in particular from solids in suspension. A little care in this direction will make all the difference in the wear on the valve and seat, and also in the life

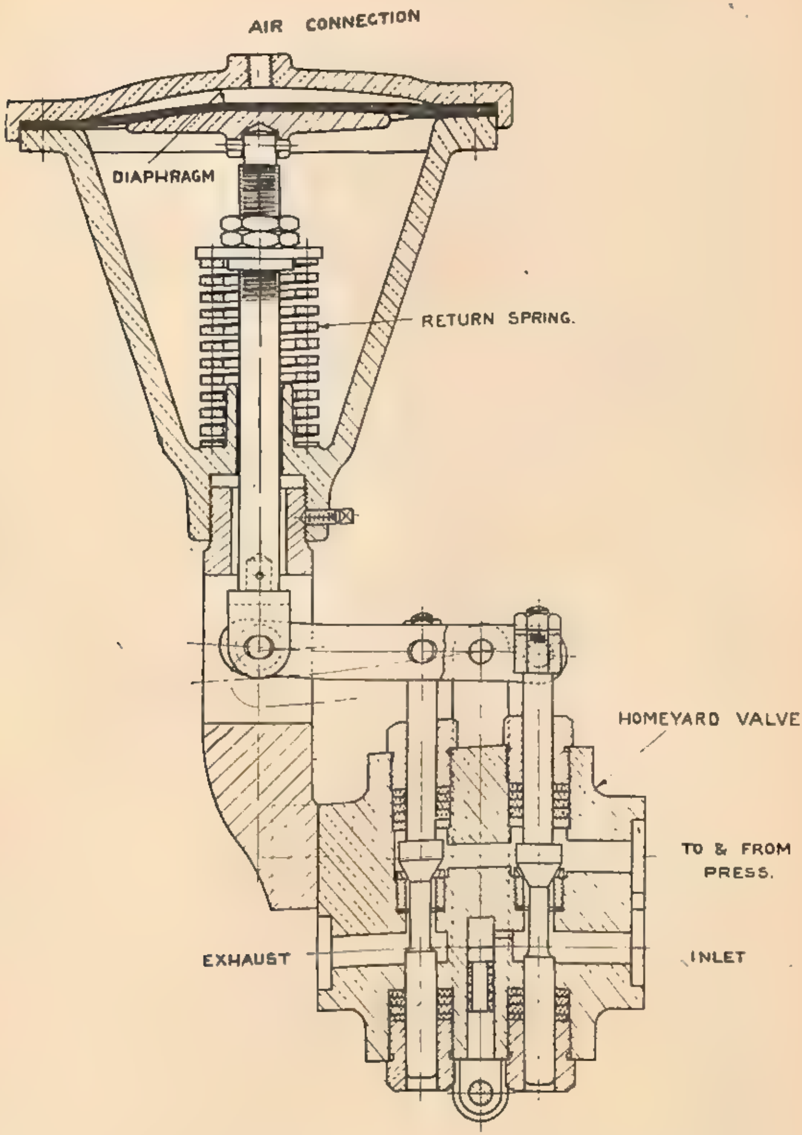


Fig. 18.—Single Ported Homeyard Hydraulic Control Valve operated by Compressed Air Diaphragm.

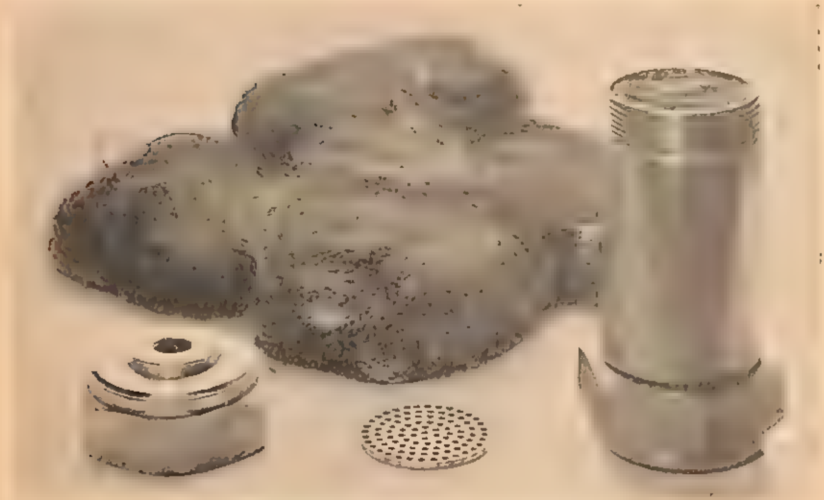


Fig. 19.—Entanglement Filter.

Glenfield & Kennedy.

of the valve in general. Fig. 19 shows a photograph of a filter which has been developed in connection with this valve and its use can be of considerable assistance in this direction.

The "Homeyard" valve is made in a series of standard sizes from ½″ diameter to 3″ for the single and from ½″ to 2″ for the double type; and, of course, for special applications and special pressures variations of the standard type can be made.

Section 4.

SLIDE AND PISTON VALVES.

The slide and piston valves constitute the remaining types in general use in high-pressure hydraulic work, and because they are somewhat similar in their method of operation they have been included together in this section.

A short reference to the slide valve was made in Section 2, when it was stated that the slide valve was one of the oldest types in use in hydraulic work. As the rotary face valve has largely replaced it, having proved more efficient, it is not proposed to devote much space to a description. In essence the slide valve is a development of its steam prototype and consists of a central chamber through which runs a rod somewhat similar to the piston in the piston valve

(see Fig. 20). Attached to this is a gunmetal slipper and, when the rod slides backward and forward, it moves the slipper across the valve seat, opening and closing the ports as required. The slipper is held against the seat in the same fashion as the rotary face valve, *i.e.*, by hydraulic pressure. The friction in this type is greater than that in the rotary face valve, and it suffers from the additional disadvantage that it is not balanced. Due to this friction the life of the valve is shorter and, generally speaking, it is only found on the older installations, and is gradually being replaced by some other more suitable type.

Although the piston valve is also one of the oldest types in use in hydraulic systems, it is worthy of note not only because it is

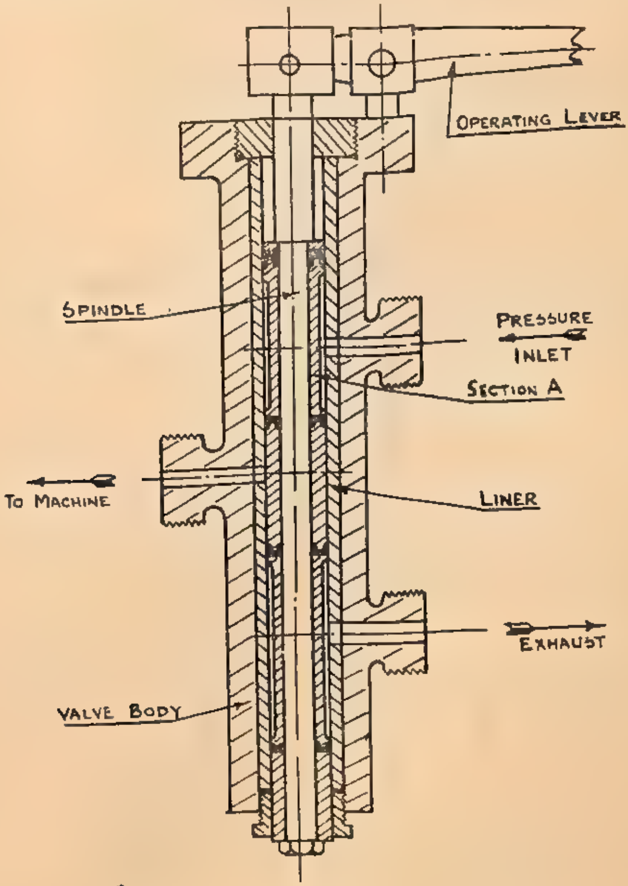


Fig. 20.—Typical Section of Piston Valve.

still widely used, but also because recent developments in the closed system with an individual pump for each machine have given the piston valve a new and increasing importance.

A typical piston valve for use in the open hydraulic system is shown in Fig. 20. It is of the single-acting type and has the usual three ports—"to machine," "inlet," and "exhaust." It consists of a central rod on which the piston sections are mounted, with leathers between them, and the whole assembly is screwed up tight by means of a nut working on a screwed portion of the end of the rod. This piston assembly slides inside a liner, which is inserted into the body of the valve and held and made watertight by means of glands and packing. This liner has ports drilled in it opposite the ports in the body, and as a large number of ports can be provided around the circumference of the liner it is possible for this type of valve to pass a larger quantity of water with less head loss than any other type of valve. In the figure the valve is shown in the central position, when all ports are closed, and it can be seen that in this position there is a leather effectively isolating each port from the other. If, however, the piston is moved downwards, a section which is turned to a smaller diameter (section A) will move until it is opposite both the pressure and the top "to machine" port, thus allowing the high pressure water to pass through to the machine while a leather still seals the exhaust port. When the piston is moved fully upwards, however, the reverse happens and the exhaust port is now connected to the machine while a second leather seals off the high pressure water.

The type of piston valve described above is most suited to the older type of open hydraulic system where pressures up to about 1500 lb./sq. inch are employed and ram sizes are larger, necessitating the handling of large quantities of water. At higher pressures the advantages are less and the question of the increased packing friction introduces difficulties. There is also a tendency for the valve to "kick," due to the fact that the turned-down sections tend to make the valve slightly unbalanced at the positions where the leathers pass over the ports in the liner, and this naturally is accentuated at the higher pressures. Wear on the leathers also becomes excessive at the higher pressures, although the valve has the advantage of being easily accessible and the fitting of new leathers is quite a simple operation. On low pressures, however, the valve is reliable and effective and will work for considerable periods without attention.

It is on the closed hydraulic system that the piston valve is assuming a new and increasing importance, and here it is being used not only as an operating valve but for many other purposes as well. The possibility of this new development in the use of the valve has come about through the fact that in a closed system where only a small amount of fluid is required it is possible to use

a special high grade oil as a medium and at the same time it is relatively easy to ensure that it is kept perfectly clean and free from any solid impurities, and also from dissolved air or water. Under these conditions the conventional forms of packing which are necessary in the open hydraulic system become redundant and freedom from leakage can be obtained at pressures as high as three tons per square inch. It should be noted that in this type of equipment the production methods are completely different from those used in the open system type of hydraulic engineering; the use of oil as a medium permits the use of hardened alloy steel bodies and liners with glass hard steel spindles (or plungers in the case of the pumps) lapped into them with clearances which are measured in ten-thousandths of an inch.

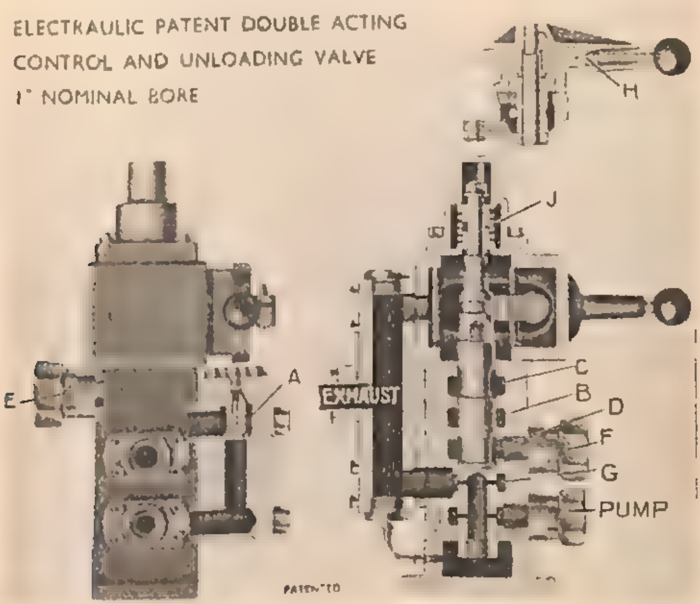


Fig. 21.—"Fingrip" Double-Acting Control and Unloading Valve. One inch nominal bore. (*Towler*).

Fig. 21 shows a sectional view of the "Fingrip" patent double-acting control and unloading valve, which is an excellent example of this trend of development of the piston type of valve. It will at once be obvious that, with the removal of the necessity for packing, the clearances between the various ports can be reduced to fractions of an inch, thus permitting of an arrangement which would be impossible had it been necessary to allow space for packing. At the same time the problem of increased friction with an increase

of pressure does not arise and the valve can be made perfectly balanced at all positions. In the example shown, the valve is balanced axially by the admission of exhaust pressure to both ends of the piston spindle, while lateral balance is ensured by annular chambers completely surrounding the piston spindle at all points where the transfer of pressure liquid takes place.

From the description of the double-acting and by-pass rotary face valve (Fig. 7) given in Section 2, the reader will be familiar with the principles of operation of this type of valve and will recognise connections F (connected to chamber D) and E (which is connected to chamber C) as the "to machine" connections, while the exhaust connection and the pressure inlet from the pump are marked in the figure. In this valve, however, there is no blanked-off position and in the centre position, in which the valve is shown, the pump is unloaded through the by-pass port at G, while the pressure is locked in the press cylinder. By moving the operating lever up or down pressure can be supplied to the appropriate machine port, while the other is opened to exhaust. In these positions when the press ram comes to a stop, the pump discharges through a relief valve or an automatic unloading valve, both of which are described below. There are, however, two intermediate positions at which it is possible partially to unload the pump and at the same time to provide a restricted flow to either end of the ram and thus move it slowly up or down. It is of interest to note that between the pressure inlet and chamber B which supplies the machine ports, a non-return valve A is fitted in order to enable the pressure to be locked in the cylinder, and another point of interest is that when the twist grip arrangement H is released, the spring J tends to restore the valve spindle to the centre position in which the pump is unloaded.

In considering any closed hydraulic system which uses an individual pump for each machine, it will be clear that the greatest care must be taken to prevent the possibility of the hydraulic circuit being suddenly broken. If, as is generally the case with high-pressure equipment, the constant delivery type of pump is used, it is clear that an open circuit condition would cause a pressure to be built up which would lock the pump and cause serious damage. For this reason the provision of a suitable relief valve or alternatively of some efficient form of unloading gear assumes paramount importance.

Fig. 22 shows a photograph of a relief valve which has been specially designed for use in this type of system. Its general action is similar to that of the mushroom relief valve described in the previous section, but it has some modifications which are of considerable importance. As it is designed for use in a closed system it is of the packingless piston type and thus has all the advantages mentioned in the description of the control valve above.

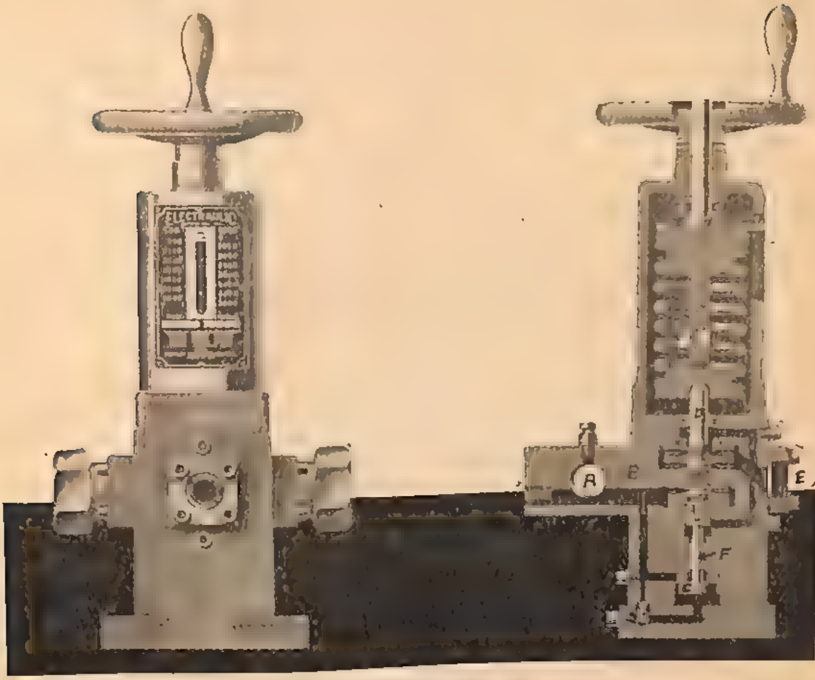


Fig. 22.—Electraulic Silent Relief Valve. (*Towler*).

The function of the accumulator in the open system is not only to store water under pressure against the varying demands of the machines, but also to help to smooth out the pulsations caused by the pump. In the closed hydraulic system there is, of course, no accumulator ; and, as mentioned before, the pumps run at far higher speeds. When these pump pulsations increase to a frequency of several thousand per minute their effect can be much more serious, and damaging vibrations can easily be set up. If, under these conditions, the relief valve should begin to hunt, or oscillate (*i.e.*, open and close with the pulsations) the vibration would be amplified to an intolerable extent. To prevent this, the "Electraulic" silent relief valve has a damping device incorporated. At the base of the valve there is a special damping piston C. It should be noted that this piston acts on the dashpot principle, pressure fluid from port A, which is in the main pressure line, going through port B directly to the top, and through a non-return ball valve to the bottom of the piston. The effective force exerted against the spring is thus the pressure in lb./sq. in., multiplied by the area in square inches of the spindle F. As soon as the pressure in the main line increases sufficiently for this force to overcome the

reaction of the spring the valve opens to the position shown in the figure and the pressure fluid is enabled to flow through port B to the annular groove in spindle D and thence through a hole in the centre of this spindle to similar grooves opposite the exhaust port E. When, however, the pressure at A falls, the valve is not able to close immediately, due to the fact that the fluid in the chamber underneath piston C is trapped by the closing of the ball valve and has to leak away past piston C before the valve can close. This damping device effectively prevents any hunting throughout the wide range of pressures at which the valve can be set to blow off. This variation is, of course, achieved by adjusting the handwheel until the spring is compressed and thus preloaded to the required extent. The two branches from exhaust port E leading to the top and bottom of spindle D are bleeder ports to prevent any pressure being built up in either of these places, due to slight leakage which may come from the pressure side of the valve.

A final point of interest concerning this valve is that, due to the absence of packing friction the accuracy of calibration can be maintained at a high level throughout the life of the valve.

Any reader who wishes to obtain more information regarding pulsation effects and the general design of high-speed pumps will find the paper to the Institution of Mechanical Engineers, "Recent Developments in High-Speed Reciprocating Pumps," by F. H. Towler, M.I.Mech.E. and J. M. Towler, of considerable interest.

While the relief valve thus makes adequate provision for the needs of the constant delivery pump in most ordinary work, there have been recent developments, particularly in the plastic and rubber industries, which impose even more stringent conditions on the press gear which is used. In particular, there are curing processes which necessitate lengthy periods of curing—in other words, the press must remain down exerting the full pressure for a very considerable period. In a case such as this it is clear that while the relief valve would certainly work, the power required to keep it lifted against the pressure of the spring would all be wasted in heat, and apart from this wasted power, the attendant troubles, due to aeration, sludging and deterioration of the oil, make it a rather unsatisfactory method. At the same time it is highly desirable that the pressure exerted by the press during this period, while remaining constant for any particular operation should be capable of being varied over a wide range to suit other types of similar curing work which the same press may be required to perform.

Figure 23 shows an automatic unloading valve which fulfils these difficult conditions very well. It completely unloads the pump except for short intervals when any leakage is being made up, and at the same time the pressure on the ram is maintained within the limits of $\pm 5\%$. It is entirely automatic and should

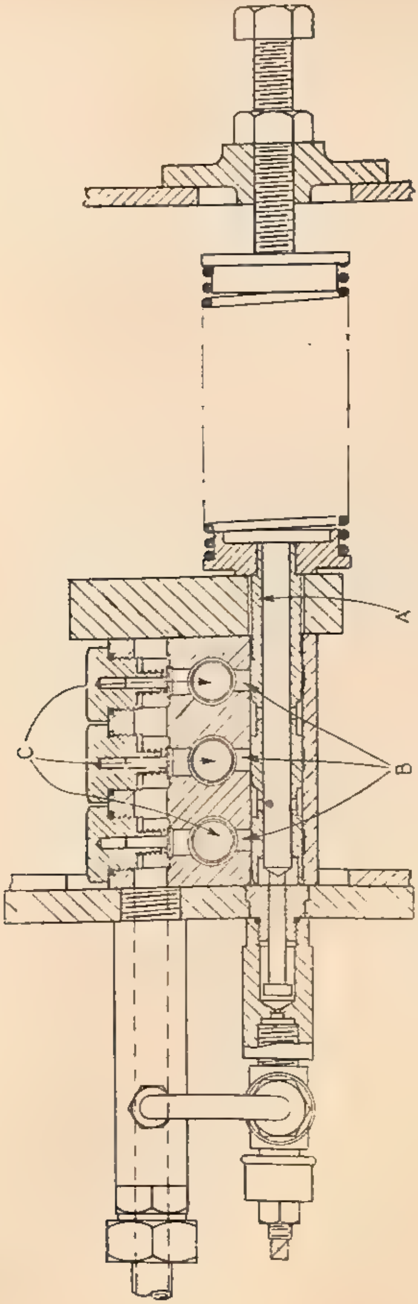


Fig. 23.—Section of Automatic Unloading Valve. *Towler Bros. (Patents) Ltd.*

the full supply of pressure be required at any time it is immediately available.

On examination of the figure it will be seen that this unloading valve is built as an integral part of the patent "Electraulic" sustained pressure pump, a photograph of which is shown in Fig. 23A. Again, the valve is of the packingless piston type, the piston being lapped and finished in the manner described above. The three pump chambers C of the three-throw pump are connected to the valve through the ports B, while one end of the piston is fitted to a spring which can be adjusted so that it balances the thrust exerted by the high pressure fluid on the other end of the piston. In the figure the valve is shown in the fully-loaded position, and in this case the pump acts as a normal high pressure constant delivery pump. When, however, the ram of the press has travelled down

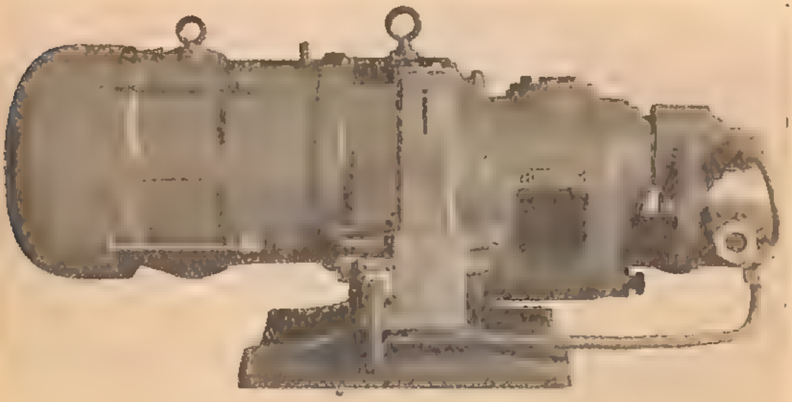


Fig. 23a.—"Electraulic" Sustained Pressure Pump, with a capacity of 2.9 galls. per minute, at 2 tons per sq. in. (*Towler*).

and is exerting the full pressure on the work, the 5% increase on the high pressure side of the pump overcomes the reaction of the spring and moves the piston across, and the three ports B are connected through the annular grooves in the piston direct to the reservoir in which the whole gear is immersed and thus the pump is completely unloaded. The pressure in the delivery side of the pump is, of course, held by the mushroom type delivery valves of the pump itself. On close examination of the figure it will be seen that the annular grooves in the piston are so arranged that the pump rams are unloaded in sequence as the piston moves across from left to right. This refinement is very effective in preventing any shock to the system which might be caused by a sudden and complete loading or unloading of the pump.

Although it is the intention of the writers strictly to confine themselves to control gear, it is necessary to mention in connection

with the closed hydraulic system that in many cases provision is made for a large capacity low pressure pump as well as the small capacity high pressure pump mentioned above. This effects a considerable saving of power on the idle stroke of the press and at the same time greatly increases its speed. When the ram reaches the work and the pressure rises, the low pressure pump is cut out and the high pressure pump automatically takes over to complete the working stroke. The low pressure pump can be of quite simple construction and is usually of the geared type.

Section 5.

THE HYDRAULIC ACCUMULATOR.

In some of the previous sections, reference has been made to accumulators and their uses in the centrally-supplied hydraulic system. As the accumulator is an important part of the control gear it is proposed to discuss it at some length and to give a description of the more important types in general use.

The main function of a hydraulic accumulator is to act as a reservoir for the high pressure water and thus to provide a balance between the supply from the constant delivery pumps and the intermittent demands of the various machines. In addition, the accumulator, if properly designed, assists in smoothing out the pulsations (which in this system are comparatively slow) from the pumps. On the other hand, a badly designed or badly situated accumulator arrangement may actually amplify the pulsations and give rise to harmful surge effects in the system.

Another point worthy of consideration is the fact that, since the accumulator requires packing and glands in the same fashion as other hydraulic equipment, there must inevitably be a certain amount of friction—sometimes absorbing up to 10% of the total power—between the ram and the cylinder. This friction will act in such a manner as to increase the water pressure required when the accumulator is being filled and will cause a corresponding drop in the pressure as it is being emptied. Although this pressure variation does not materially affect the working of the normal open hydraulic system, it is a point which must be taken into account in the design of the machinery, relief valves, etc., used. The speed at which the accumulator rises and falls materially affects both the normal variation in pressure and the surge pressures which may be set up, and for this reason it is designed so that the speed when falling will not exceed about 12 inches per second under the condition of maximum demand.

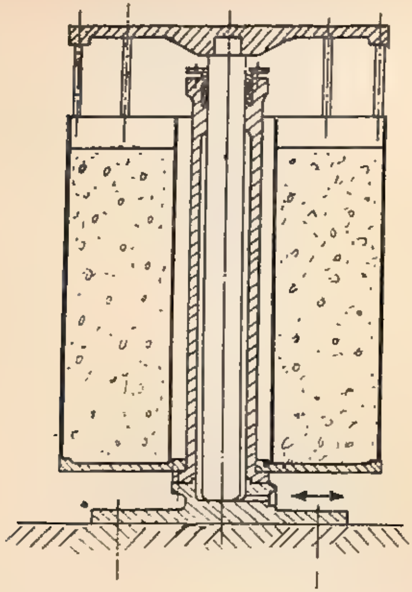


Fig. 24.—Section of Hydraulic Accumulator.

Figure 24 shows a section giving the salient features of the usual type of weight loaded (weighted casing) hydraulic accumulator. It consists of a cylinder mounted vertically, in which acts a ram with gland and packing arranged in the same manner as in an ordinary machine. On the top of the ram is mounted a crossbar and from this is suspended either a hollow casing which is filled with slag, scrap iron or some similar heavy material (weighted casing type), or else large slabs of iron which are specially cast (weight loaded type). The latter, of course, makes for a more compact machine. Another variation which is sometimes employed is to mount the ram on the baseplate and to have the cylinder moving. There is a certain advantage about this method in so far as it is then possible to bolt the casing to the cylinder at the bottom as well as at the top and avoid any overhang. The gland and packing are less exposed—an important consideration if the accumulator is to work in the open—and there is less chance of grit and foreign materials working their way into the packing and scoring the ram. On the other hand the gland is naturally more difficult of access and any overhaul or repacking of the accumulator becomes a much more troublesome matter, while the regrinding of the ram entails the complete dismantling of the machine. In heavy types, this more than balances the advantages of this form of accumulator.

Whether the cylinder or the ram of the accumulator is fixed, it is clear on examination of the figure that the ram of the accumu-

lator will have to carry a very considerable weight amounting, in the larger sizes, to hundreds of tons. At the same time, the large size of the casing will cause considerable lateral stresses. These can be minimised to a certain extent by the use of external guides on the accumulator, and these also provide a means of preventing the rotation of the weighted casing and supporting gear for actuating the relief valve and/or the tappets for stopping and starting the pumps. Nevertheless the utmost care must be taken to ensure that the diameter of the ram is such that it is easily capable of supporting any stress to which it may be subjected. A usual practice is to relate the stroke of the accumulator in feet to the diameter of the ram in inches. This is not, of course, an exact rule, and considerations of the size of the casing, the wind pressures it may be subjected to, the stroke of the accumulator and the capacity and water pressure, all play a part in determining the most suitable proportions. While the minimum size of the ram is thus determined by considerations of its mechanical strength, the maximum size is mainly determined by the allowable loss due to friction in the packing ; for this increases in an amount roughly proportional to the circumference of the ram ; and an excess of friction will, as explained above, result in an excessive change in the water pressure in the system.

In some special cases, when it is required to store water at an unusually high pressure, it may be found that the size of the ram required is too small for the considerations of strength mentioned above. In this case an interesting variation of the normal accumulator, known as the differential type, may be employed.

Fig. 25 shows an example of this type of accumulator. It will be seen that in this case the ram is extended so that a portion—slightly less in diameter—is projecting through the top of the cylinder. This projecting portion must, of course, be of sufficient length to accommodate the cylinder for the full length of its stroke. The differential accumulator takes its name from the fact that in this case the effective area is the difference between the areas of the bottom and the extended portion of the ram and is not related to their actual diameters. This relationship also holds good for the capacity of the accumulator.

While at first sight the differential accumulator appears to have certain definite advantages, it possesses some practical difficulties which must be carefully considered. While it has the undoubted advantage that the top and bottom bearings on the ram make for increased stability, the necessity for providing an extra packing somewhat increases the cost of construction and maintenance, and in addition the increase of packing friction constitutes a serious problem. Here it must be remembered that owing to the differential action a much lighter load is required and as the packing friction is mainly proportional to the circumference of the ram and

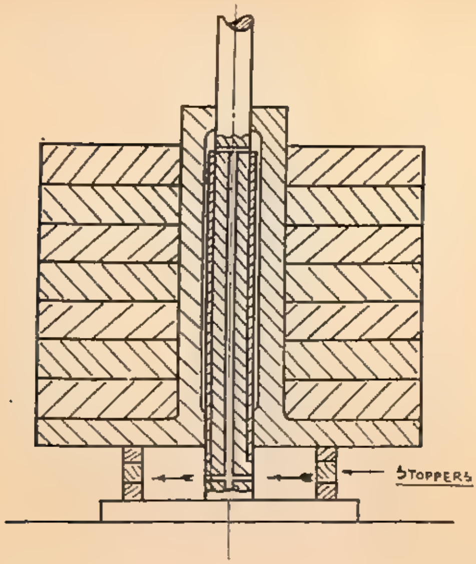


Fig. 25.—Hydraulic Accumulator—Differential Type.

the water pressure, it will constitute a much larger proportion of the total load and will tend to create a greater change in pressure in the system than is allowable.

In conclusion it is interesting to note that while a flexible connection to the moving cylinder type of accumulator is possible, it would be very cumbersome in practice and is never used. Instead it is customary to drill a hole through the centre of the ram in the cases where it is fixed to the baseplate and to provide fixed connections at the base through which the water enters and leaves the accumulator.

The following typical calculations are intended to cover some of the more important points mentioned above :—

Problem.—It is desired to design an accumulator having a capacity of 30 gallons and suitable for a working pressure of 1500 lb./sq. inch.

As this is a normal pressure, the normal type would be used. It is not possible to discuss the more intricate details of design, but it can be stated that for most conditions either the moving ram or the moving cylinder type could be used.

As stated above, a good basis for estimating the proportions is to assume that the diameter of the ram in inches will be roughly equal to the stroke in feet, and from the table in the appendix we can work out the following sizes :—

Ram Dia.	Galls./Ft.	Stroke (Feet).	Load (Tons).
8″	2·171	13·8	34
9″	2·747	10·9	43
10″	3·393	8·9	53

The stroke is derived from the simple formula :

$$\text{Stroke in Feet} = \frac{\text{Capacity of Accumulator (Galls.)}}{\text{Equiv. Galls./Ft. of Ram Diam.}}$$

and the loading required for the pressure is

$$\text{Load in tons} = \frac{\text{Ram Area (Ins.)} \times \text{Water Pressure (lbs./sq. in.)}}{2240}$$

From the above table it can be seen that a suitable size for the accumulator would be 9″ diam. ram × 11 ft. stroke.

Having determined a provisional size, the following are the more important points it is necessary to check.

Fig. 26.—Homeyard Valve—Solenoid-Operated.

Glenfield & Kennedy.

Thickness of Cylinder.

Making a nominal allowance for clearance of 1″ each side, we have the internal diameter of the cylinder set at 11″. Having determined this, the Lamé thick cylinder formula can be used.

This states :

$$t = d/2 \left(\sqrt{\frac{f + p}{f - p}} - 1 \right)$$

where

- t is the thickness of the cylinder walls in inches.
- d is the inside diameter of the cylinder in inches.
- f is the allowable hoop stress in lb./sq. inch.
- p is the water pressure in lb./sq. in.

Using a high duty cast iron, we can take the stress as 5000 lb./sq. inch. Thus :

$$t = \frac{11}{2} \left(\sqrt{\frac{5000+1500}{5000-1500}} - 1 \right)$$
$$= 5{\cdot}5\ (\sqrt{1{\cdot}85} - 1)$$
$$= 5{\cdot}5 \times 0{\cdot}36$$
$$= 1{\cdot}98'' \quad \text{say } 2''$$

Thus the cylinder will be 11″ *i/d* × 15″ *o/d*.

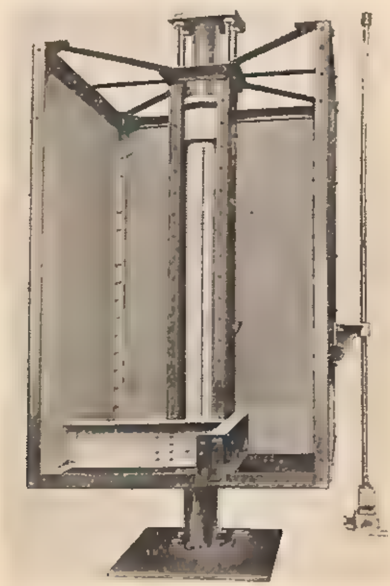


Fig. 27.—Typical Large Accumulator.
Hugh Smith & Co. (Possil) Ltd.

The Baseplate.

In this class of work it is usual to allow a bearing pressure on the foundation of about 2 to 5 tons/sq. ft.

Assuming in this case that the pressure is to be 2 tons/sq. ft., we have

Load on baseplate, 43 tons (from table) + Allowance for cylinder, etc. (assume this amounts to 7 tons).

$$\text{Total} \doteq 50 \text{ tons.}$$

$$\text{Required area} = \frac{\text{Total Load}}{\text{Pressure (tons/sq. ft.)}}$$

$$= \frac{50}{2} = 25 \text{ sq. ft.}$$

A baseplate, 5 ft. square, would fulfil this condition, and at the same time would be in reasonable proportion to the remainder of the accumulator.

Strength of the Ram.—This is normally checked by treating the ram as a loaded pillar and is quite straightforward and need not be dealt with here. Two points, however, are worth noting—first, that a generous factor of safety (about 8-10) should be allowed and second, that the possibility of the accumulator, being subjected to wind stresses, should be considered and allowance made accordingly.

Size of Piping.

As the speed of travel of the accumulator must not exceed 1 ft./sec., it is clear from the table in the appendix that this is equivalent to 2·747 galls./sec. If the rate is to exceed this, it is, of course, necessary to have a larger diameter ram (if friction considerations will allow) or alternatively to have a number of accumulators. This question involves consideration of the system as a whole and relates to detailed design considerations rather than general principles.

Assuming this capacity (*i.e.* 2·747 galls./sec.) is adequate, we have

$$\underset{\text{(Ft./Sec.)}}{\text{Speed of Flow in Pipe}} = \frac{\text{Acc. Capacity (galls./sec.)}}{\text{Galls./ft. for Pipe}}$$

As has been stated before, the speed of the water through the pipes should not exceed about 10 ft./sec.

Thus the galls./ft. of the pipe should be

$$2{\cdot}747/10 = {\cdot}275 \text{ galls./ft.}$$

From the table, we see that a 3″ diameter pipe, with a capacity of ·305 galls./ft. would satisfy this condition.

Air Hydraulic Accumulators.

Another form of accumulator which is growing in importance for certain types of hydraulic installation is the air-hydraulic accumulator, and Fig. 28 gives a diagrammatic outline of a typical installation. The air-loaded or air-hydraulic type of accumulator takes its name from the fact that compressed air instead of weights is used as a balancing medium. It consists essentially of a fixed ram on the water side with a moving ram between this and the fixed air cylinder. The supply of air to the cylinder is maintained by a small compressor and is stored in one or more air bottles as shown. The compressor unit is small and takes from 8 to 48 hours fully to charge the installation and thereafter is used for the purposes of maintaining the pressure. A short run once a week may be sufficient.

It should be noted that the air and water are kept quite separate, and this distinguishes the air-loaded accumulator from the hydro-pneumatic type, where the air and the water have a common surface.

Since, assuming isothermal expansion $P1\ V1 = P2\ V2$, it will be clear that if the variation in water pressure (which is directly proportional to the variation in air pressure) is to be maintained between 5%-10% above or below normal, the water capacity of the installation will be proportional to the air capacity of the cylinder, air bottles, etc. The pressure at which the air is maintained also affects the total volume of air required since a higher pressure means that the ram of the air cylinder can be proportionately smaller. For hydraulic systems, using a water pressure of 1000 lb./sq. in. and upwards, an air pressure ranging from 600-800 lb. is usual. At this pressure it is necessary to take the greatest care in the construction of the air bottles. These are made from solid drawn tube and must meet B.O.T. safety requirements.

The saving in weight is one of the outstanding advantages of the air-loaded accumulator and this makes for cheaper foundations and requires less floor space. At the same time the system is more easily dismantled when repairs are necessary; and the lightness of the moving ram has a very great effect in reducing the shock pressures which are so liable to be built up in a system which uses a weight-loaded accumulator. From this it follows that the allowable speed of travel of the ram can be considerably greater in the air-loaded type of accumulator. An additional advantage is that it is comparatively easy to vary the pressure in the hydraulic system should this be necessary.

Although the air-loaded accumulator requires a compressor and air bottles which are not necessary with the normal accumulator its great advantages, particularly in larger systems, are giving it a wide and ever-increasing popularity.

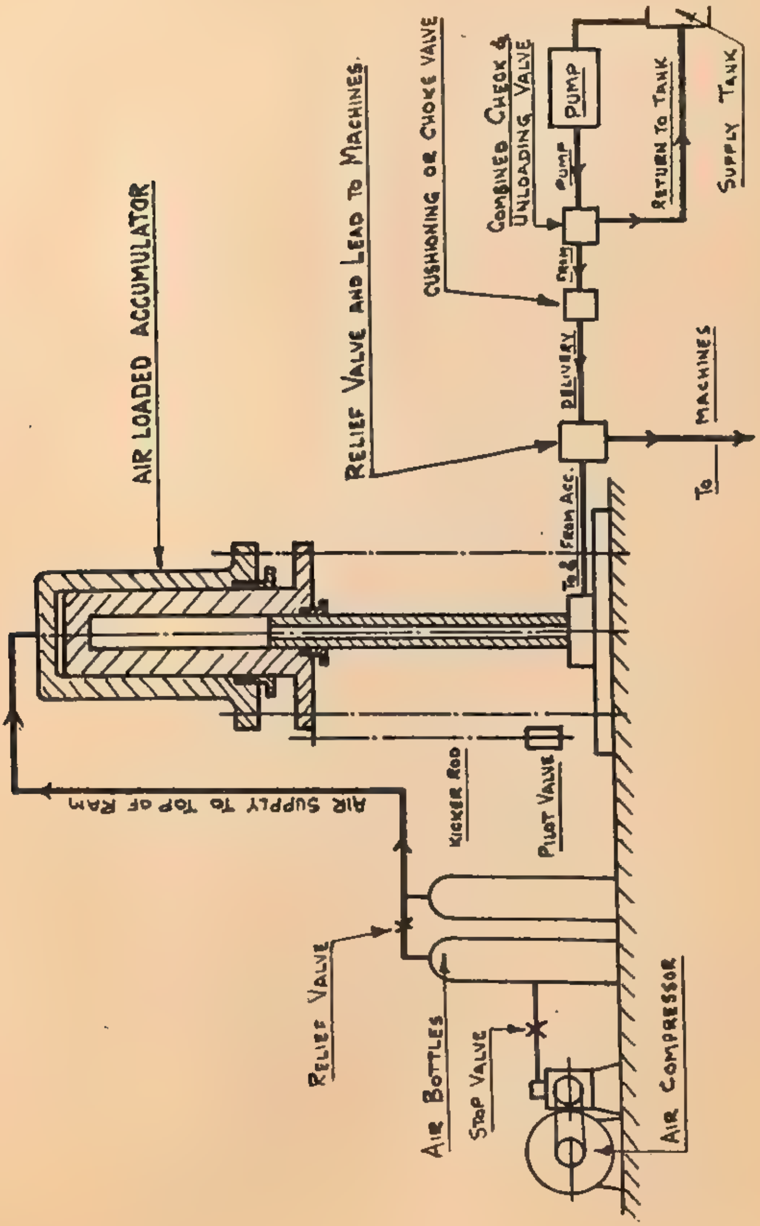


Fig. 28. -Diagrammatic Layout of System, with Air-Loaded Accumulator.

Section 6.

SPECIAL HYDRAULIC ARRANGEMENTS.

Although most of the various types of control valves in common use have now been discussed, there remains one important aspect of control gear which has yet to be mentioned. This is the use of combinations of valves and other gear to produce results which cannot be achieved by any one valve alone.

A simple illustration of the principles involved is shown in Fig. 29. This is a diagrammatic sketch of an arrangement for operating a large hydraulic press used in an open system. In connection with this, it will be remembered that while the rotary

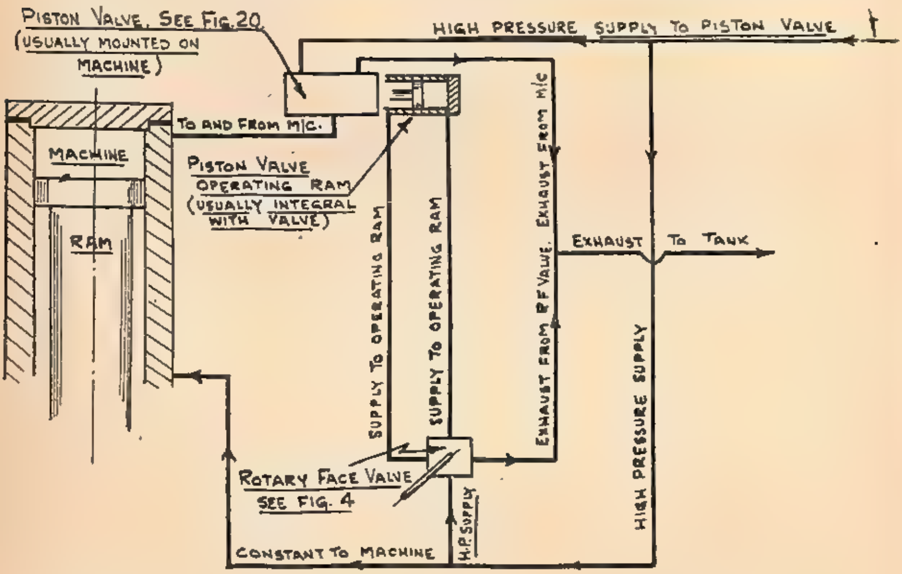


Fig. 29.—Diagram of Valves and Connections for Large Machine Operation.

face valve has many advantages, it is limited in use by the relatively small amount of water it can pass. On the other hand, the piston valve is very useful in this respect, but is somewhat difficult to operate manually, especially in the larger sizes and higher pressures. Thus we see that if the piston valve is mounted as shown, and operated by a small ram, which is in turn controlled by the rotary face valve, we have an arrangement possessing the advantages of both types without their limitations.

It should be noted here that although the piston type valve in the open system has certain disadvantages as a simple operating valve, some form of it, actuated by a small ram, is used in a great number of these special arrangements. The "kick" effect, which

is noticeable when it is hand-operated, has no effect on its action, and a very considerable advantage arises from the fact that the piston of the valve and the operating ram can be integral. Thus there is only one moving part, and no links or other forms of connection are required.

One of the most important uses to which these special arrangements are put is the provision of some means of saving power, and some typical methods of achieving this are given below. First, however, it may be useful to explain exactly what is meant by this expression.

The term "saving power" has a special meaning in hydraulic practice, and in order to understand its significance it must be realised that each cubic foot, for example, of high pressure water represents a definite amount of stored potential energy. If this high pressure water is fed to the cylinder of a machine during the idle period of the stroke it is clear that, as far as useful work is concerned, the energy is entirely wasted. In actual fact, it is transformed into heat, generated by the great speeds and excessive agitation to which the water is subjected. In addition, the erosive effect on the ports is tremendously increased and altogether such a condition is highly undesirable. Power saving is, in essence, some method of eliminating or at least mitigating this condition, and the devices themselves are usually based on the intensifier, prefilling or slack water principles.

Fig. 30 shows a diagrammatic arrangement of an intensifier. As can be seen from the figure, it consists of two cylinders, one fixed and one moving, and a fixed ram. The hydraulic main pressure is fed to the fixed cylinder, tending to push the moving cylinder, which is also a ram, upwards. This cylinder moves up the fixed ram and will only be prevented from so doing when the pressure of the intensified water inside the moving cylinder rises to a pressure which, when acting on the smaller area of the fixed ram, will exert a force equal to that exerted by the hydraulic main pressure on the larger area of the underside of the moving cylinder.

In other words, the ratio of the main pressure to the intensified pressure will be equal—ignoring friction losses—to the ratio of the area of the fixed ram to the area of the underside of the moving cylinder. These proportions can be arranged to suit the particular requirements of the system.

Fig. 31 gives an idea of how the intensifier can be arranged as a power-saving device. The operating valve A has two open positions and when turned to the first position it connects the top of the machine cylinder via pipe B to the hydraulic main, and the cylinder fills as the ram moves down until it reaches the work. At this point the operating valve is put over to the "intensified" position. This closes the port to pipe B and at the same time opens a port connecting the low pressure side of the intensifier

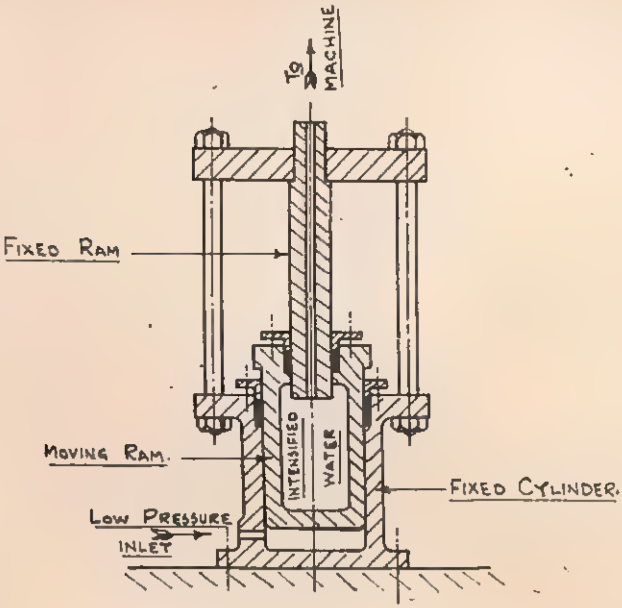


Fig. 30.—Hydraulic Intensifier.

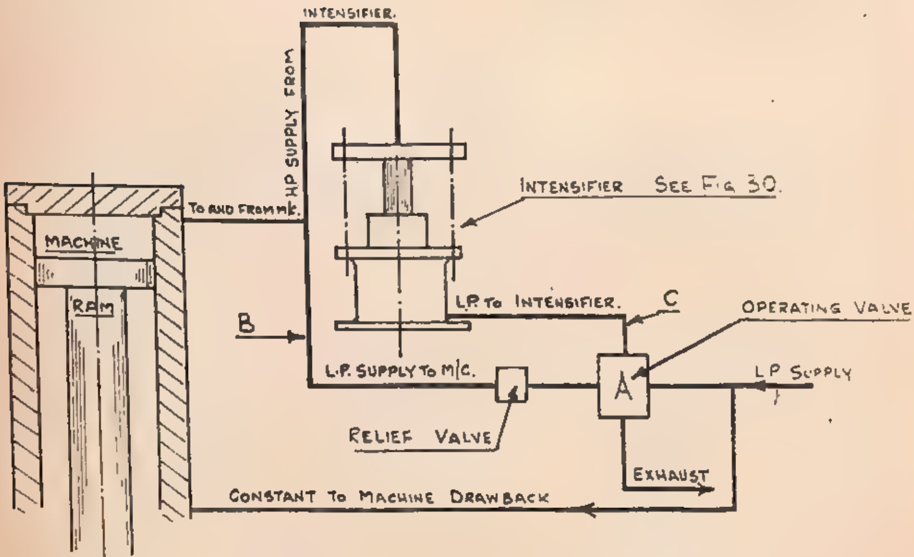


Fig. 31.—Diagrammatic Layout of Intensifier Power Saving Arrangement.

via pipe C to the hydraulic mains. As the high pressure side of the intensifier is permanently connected to the cylinder of the machine it will automatically apply the increased pressure to the ram and the working stroke will be completed. In the "off" position both pipe B and pipe C would be connected to the exhaust, and the rams in the machine and the intensifier returned to the normal position by auxiliary rams connected permanently to the hydraulic mains, or by some other convenient means.

At first sight this method may seem to be very effective, but further consideration shows that the actual power saved depends on the actual water pressures involved. From the worked example at the end of this section it will be seen that full power-saving could only be achieved if the hydraulic main were at a pressure so low that it would just be sufficient to overcome the packing friction of the machine. Needless to add, such a low pressure would not be an economical or practical proposition.

This type of intensifier arrangement, however, can prove very useful in the older type of open hydraulic system where it will not only provide a reasonable amount of power saving, but will also permit of the use of more modern machines using a higher pressure than the system is able to supply.

It is also of interest to note that with this type of arrangement it is possible to supply a machine with several different pressures from the one system; and this is useful if a number of jobs, requiring definite and different loadings, have to be done on the one machine. The machine would, of course, have to be capable of withstanding the maximum intensified pressure supplied.

The prefilling method of power-saving is commonly employed in the closed hydraulic system and is automatic, simple and effective. Fig. 32 shows a typical diagrammatic arrangement. In this case two pumps, a high pressure small delivery unit and a low pressure large delivery unit are used. The combined operating valve, usually of the piston type is so arranged that during the idle stroke the low pressure pump feeds the machine, while the high pressure pump is unloaded. When the ram of the machine meets resistance and the pressure in the supply line rises, the increased pressure pushes the piston valve over against the reaction of a spring, and the low pressure pump is then unloaded while the high-pressure pump automatically completes the working stroke. As in this case, the pumps and pressures can be made exactly to suit the individual machine, complete power saving can be achieved; and at the same time the idle stroke can be speeded up by the large delivery from the low pressure pump until it will equal that of any mechanical counterpart. The packingless piston type of valve and all the other advantages of the closed system can, of course, be utilised.

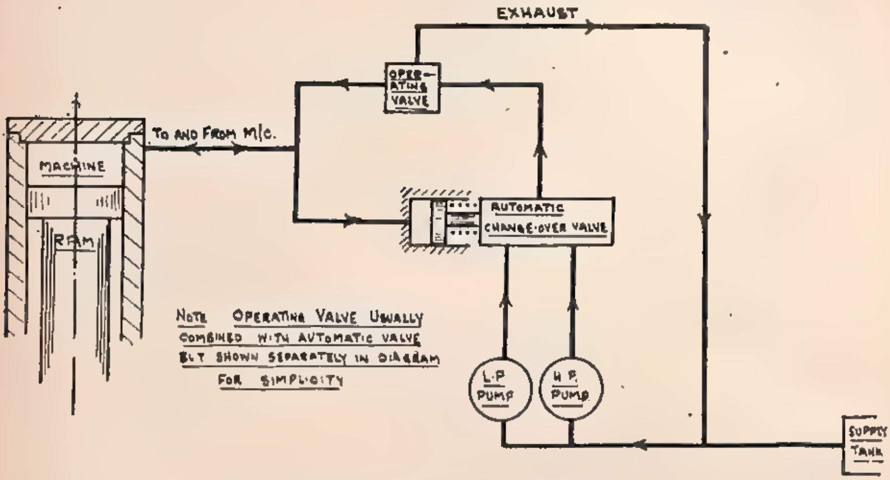


Fig. 32.—Diagrammatic Layout of Prefilling Power Saving Arrangement.

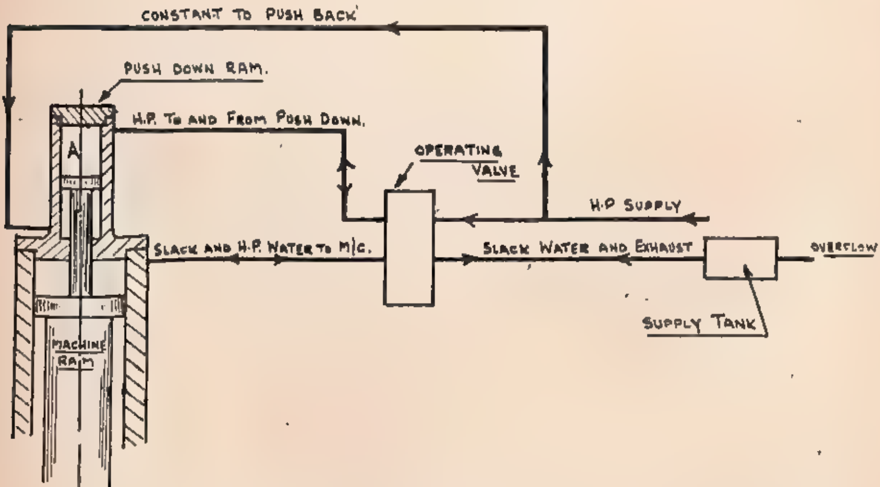


Fig. 33.—Diagrammatic Layout of Slack Water Power Saving Arrangement.

The slack water principle is the most widely used as a basis for power-saving methods in the open hydraulic system. A typical arrangement is shown in Fig. 33 and the action is comparatively simple. The push-down ram A on the top of the machine is an essential part of this method, and when the operating valve, which again can have two operating positions, is moved to the first position, it supplies pressure water to the push-down ram and at the same time connects the main cylinder to the slack-water tank in the exhaust system. As the push-down ram moves the main ram, the main cylinder fills up with water from the exhaust system until the idle stroke is completed. At this point the operating valve is put over to the "power" position. This disconnects the supply leading to the machine from the exhaust and connects it to the high-pressure mains. Then with the full pressure acting on the ram, the machine completes its working stroke in the usual way. In the "off" position both the main and push-down ram cylinders are connected to exhaust, and the rams can be returned by constant pressure in the usual way.

This method, using slack or exhaust water, is very effective on open hydraulic systems. It can achieve full power saving and is effective at all pressures. At the same time the arrangement used is relatively simple and inexpensive. The main difficulty—particularly when fitting this method of power saving to existing machines—is that of fitting the push-down ram to the machine. Although only one ram is necessary, it is usual in the larger machines to fit two or more, so that the main ram is not likely to be jammed by an uneven thrust. In small machines one push-down ram can be fitted on top of the machine on the centre line of the main ram. By varying the size of the operating valve and the push-down cylinders the speed of the idle stroke can be arranged to suit requirements up to the highest idle speeds obtained with a normally-operated machine of a similar type.

A very interesting arrangement of this type, using a "Hugh Smith" patent power-saving valve is shown in Fig. 34. This valve is entirely automatic in its operation and can be controlled by a small rotary face-type valve. The change over from slack water to high pressure water takes place automatically when the ram of the machine reaches the work, and thus the operating valve need only have an "on" and an "off" position. It is of interest to note that this power-saving valve, together with an individual slack-water tank, are mounted on top of the machine, making a neat, strong and efficient connection and effecting a considerable saving in floor space.

There are a number of processes, such as riveting, where it is necessary that the press should remain down, exerting its full pressure, for an appreciable time after the stroke has been completed. Here the hydraulic press scores heavily over any mech-

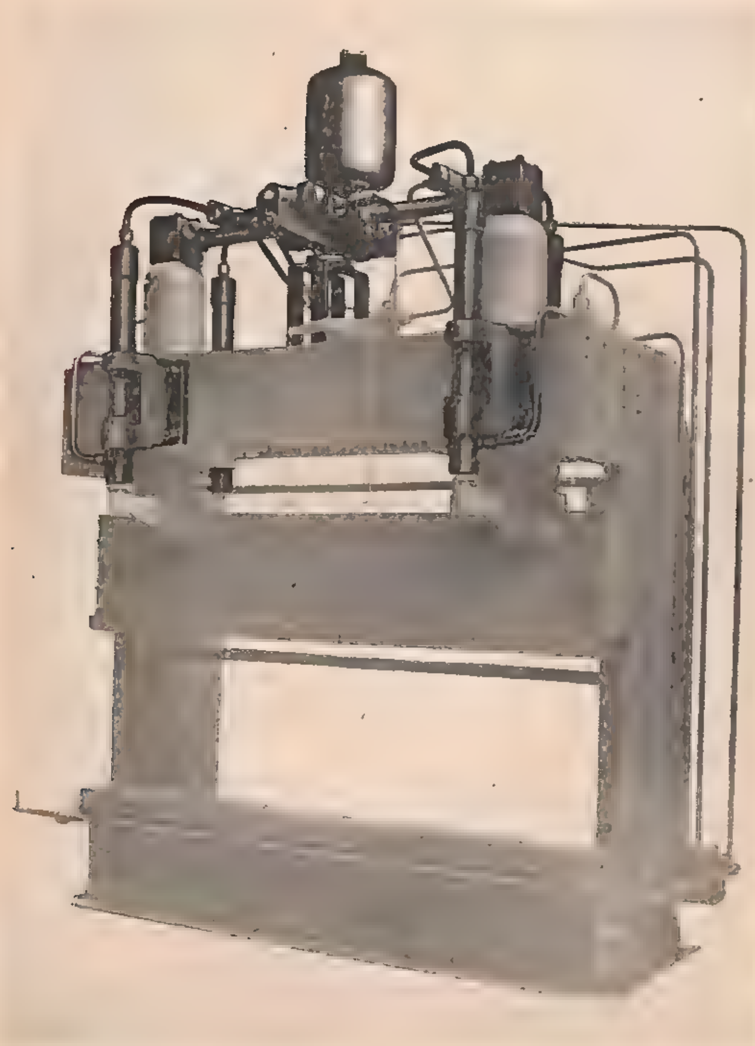


Fig. 34 illustrates a Hydraulic Flanging and Folding Press, showing the application of the Hugh Smith Patent Power Saving Equipment. The photograph clearly shows the small bore auxiliary cylinders and water reservoir. Hugh Smith Patent Power Saving Equipment is incorporated as standard in many new machines, but can also be fitted to existing hydraulic machines of almost every type.

On account of the great reduction in the consumption of power water, the installation of this equipment can frequently save the cost of the new pumps or accumulators which would otherwise be required. Operation is also greatly speeded up.

anical counterpart, as the position of the ram can be controlled by the operating valve, and will remain in any position as long as desired.

In considering the processes mentioned above, it is clear that when repetition work is involved, it is a great advantage if the time which the press remains in the "down" position can be made independent of the operator. For this purpose an arrangement using a timing valve is employed.

An illustration of a simple form of this arrangement is given in Fig. 35, and the action is quite simple. When the operating valve is opened, pressure water is supplied to pipe B and also to port A of the timing valve. Lifting the ball valve the water passes through the timing valve and moves the small cylinder which in turn opens the control valve and allows pressure water to be supplied

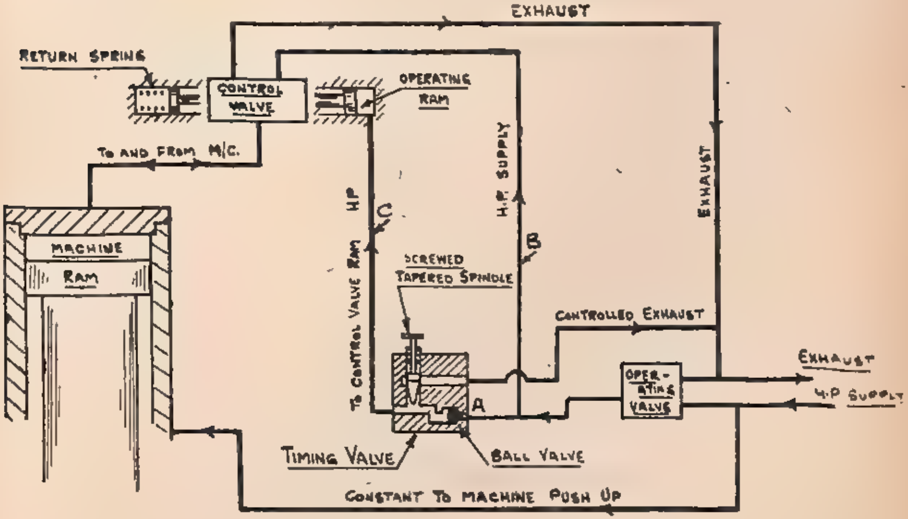


Fig. 35.—Diagrammatic Timing Arrangement.

to the machine cylinder via pipe B. When the press has completed the stroke and the operating valve is put over to exhaust, the machine ram does not, however, at once return to its top position. The closing of the ball valve in the timing valve prevents the operating ram in the small cylinder from returning and thus the control valve is held open. The pressure in this small cylinder, however, gradually falls, due to the leaking away of the water through the timing control device which can be some form of adjustable needle valve. When sufficient pressure has leaked away the operating ram will be pushed over, allowing the control valve to open and the machine ram will return to the top position in

the usual manner. It should be noted that pipe B could be connected direct to the high pressure mains, but if this were done it would be impossible to stop the machine ram once the operating valve had been put over. This is rather dangerous and the other arrangement is usually adopted.

There is another arrangement which merits some mention, and that is the hydraulic reducing valve. The arrangement is usually based on the piston type valve and a diagrammatic layout is shown in Fig. 36. The piston valve is arranged as a simple "open"-"closed" valve and at each end is an operating cylinder and ram, one larger than the other. The pressure water is connected to one side of the piston valve and also to the smaller

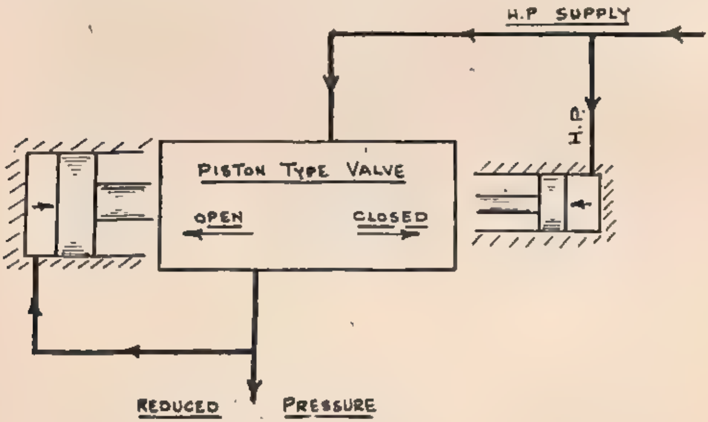


Fig. 36.—Diagrammatic Pressure Reducing Arrangement.

operating cylinder. The large cylinder and the other side of the piston valve are connected together and from this is taken the reduced pressure. When water is drawn from the reduced pressure side of the valve the pressure falls and the high-pressure operating cylinder opens the piston valve. When sufficient water has passed and the pressure rises again, the large operating cylinder closes the piston valve and the reduced pressure is stabilised.

If a spring is substituted for the high-pressure operating cylinder, it is possible to make a variable pressure reducing valve ; but it should be noted that whatever methods are employed, the reducing valve is inherently wasteful of power. Although it can be very useful in connection with small riveters, etc., it is mainly employed with hydraulic testing and similar machinery, where a variable pressure supply is required from a fixed high-pressure mains, and economy in power is not an important factor.

Power-Saving Calculations.

It is important when discussing power-saving that it should be distinguished from the question of the overall efficiency of the machine or system. While the overall efficiency is expressed by the ratio—power output in useful work/total power input, the power saving is expressed by the ratio—actual power saving/theoretical possible power saving. On examination it will be seen that this latter depends on the head losses, and is not related to questions of packing friction, etc.

The following simple example will serve to illustrate this idea. Consider a press capable of exerting a force of 100 tons and having a packing friction of 5 tons (*i.e.*, a 95-ton press). For a working stroke of 12" the work done under full load conditions would be $100 \times 12 = 1200$ inch tons, and in the normal way fluid possessing this amount of stored energy would have to be drawn from the system.

Suppose, however, the working stroke required were only 1". The theoretical possible power saving can then be calculated as follows :—

Work required for idle stroke	=	5×11	= 55 inch tons.
Work required for working stroke	=	100×1	= 100 inch tons.
		Total	= 155 inch tons.

This would represent a saving of power of 1200 - 155
= 1045 inch tons.*

Although any one of the power-saving methods could be used, it must be noted that with the intensifier method the ratio intensified pressure/normal pressure would, for full power saving, have to be 100/5, or 20/1 to 1, and such a wide variation in pressure is not a practical proposition.

In dealing with high-pressure hydraulic control gear, it has been the object of the writers to explain general principles rather than to give detailed descriptions which would have but a limited interest to the general reader. The scientific application of hydraulic methods to the realm of general engineering is, in their opinion, only just beginning ; and if the material has given the reader some idea of the possibilities, their efforts will have been successful.

* This energy, in the ordinary way, where power saving is not employed, would appear in the form of heat generated at the points where the head losses occur ; and it is interesting to note that while losses of this magnitude in ordinary machinery would at once produce local overheating, the large quantity of fluid present in the hydraulic machine will absorb and distribute the heat without any noticeable effect. For this reason the enormous losses which can occur in badly-designed hydraulic machinery are often overlooked.

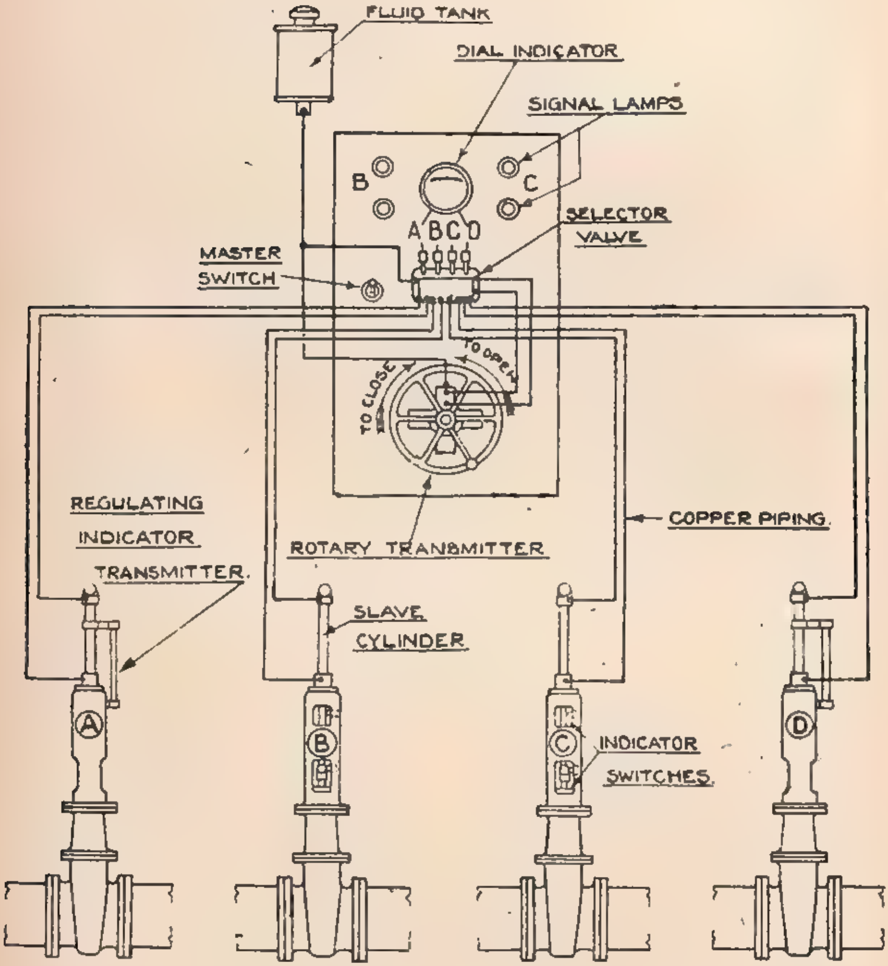


Fig. 37.—Group Operation of Valves, Lockheed Remote Control.

Glenfield & Kennedy.

Section 7.

HYDRAULIC REMOTE CONTROL GEAR.

While the subject of remote control gear is not strictly within the scope of this paper, it is a branch of hydraulic engineering of considerable importance and has several distinct advantages over electrical or mechanical remote control methods. It is independent of any external source of power and this is often a very important consideration—on board ship, for instance, when an electrical supply might easily fail at the very time when the use of remote control gear for closing bulkhead doors is most urgently required. At the same time, it is free from the cumbersome links, rods or control wires which are a feature of any mechanical system, and it can be used without the need for any provision for external lubrication. A further feature of hydraulic remote control is that the operating or slave cylinders can, by varying their size, be made to exert any force from a few pounds to several tons, irrespective of the force exerted by the operator or the pressure in the pipe lines.

The diagram given in Fig. 37, showing a typical arrangement of a remote control system, and the photograph in Fig 38 showing

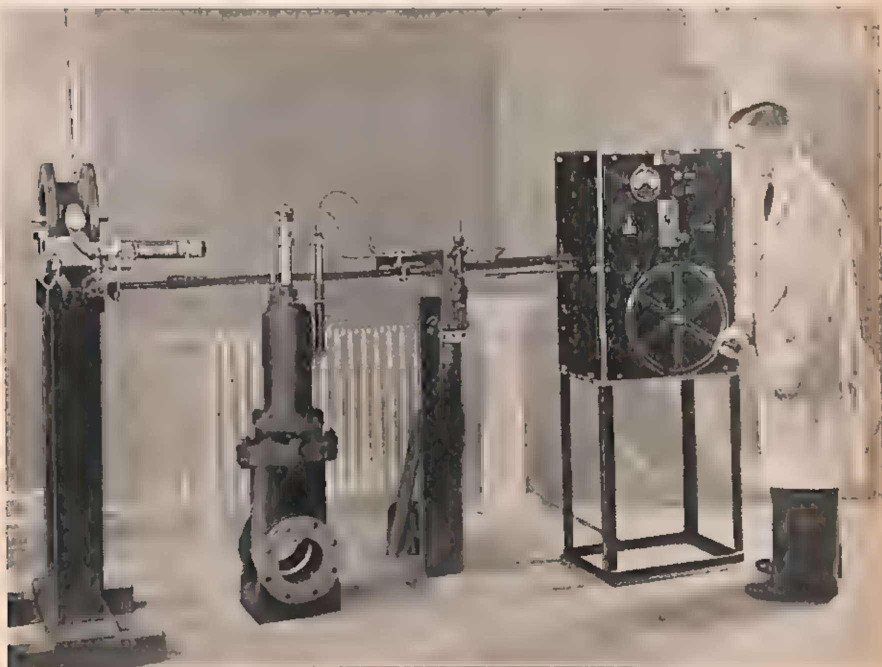


Fig. 38,—Lockheed Remote Control.
Glenfield & Kennedy.

an actual experimental layout will be of assistance to those who are not familiar with the principles of this type of equipment. It will be noticed that a system of signal lamps is provided, but this, of course, is a refinement and should the electric supply fail, the operation of the gear itself would not be in any way affected.

Essentially the system is composed of a rotary transmitter, a selector valve and the slave cylinders. Each slave cylinder is, of course, attached to the equipment it is desired to operate, and when it is connected through by the appropriate selector valve, the rotary transmitter—a special form of hand-operated high-pressure pump—can be operated to supply it with the necessary hydraulic power.

In addition to these components it is necessary to provide a a reservoir tank capable of feeding the system as required and accommodating such changes in volume as may take place due to temperature variations.

The chief qualities required in the hydraulic fluid are that it should have a high flash point, a low congealing temperature and that it should not affect or be affected by the materials with which it comes in contact.

Although the principles of hydraulic remote control are simple, there have been so many refinements and special developments that it is impossible adequately to deal with it in a work of this nature, and the writers can confidently recommend for further reading a paper, "Hydraulic Automotive Apparatus in Ships," by E. Bruce Ball, Jr., M.A., published by the Institution of Engineers and Shipbuilders in Scotland; and also a further paper, "A New System of Hydraulic Remote Operation for Control Valves" (Institution of Mechanical Engineers), by the same author.

A study of the applications and possibilities of this type of control gear is well worth while, and there is every prospect that it will play an important and useful rôle in many future developments in engineering design.

APPENDIX A.

Hydraulic Memoranda.

Weight of 1 cubic foot of fresh water = 62·4 lb.
Weight of 1 gallon of fresh water $\doteq$ 10 lb.
Column of water 1 foot high = 0·434 lb./sq. in. press.
Velocity of water (ft./sec.) = $\sqrt{2 g H}$, where

H is the head in feet,
g is the acceleration due to gravity = 32·2.

Kinetic energy of a jet of water $= \dfrac{W A V^3}{2 g}$ ft. lb./sec., where

W is the weight of 1 cub. ft. of water,
A is the area of the jet in square feet,
V is the velocity of the jet in ft./sec.
g is the acceleration due to gravity = 32·2.

Quantity discharged through small orifice = $Cd\, A \sqrt{2 g H}$ cub. ft./sec. where

Cd is the coefficient of discharge (determined experimentally).
A is the area in square feet.
H is the head in feet.
g is the acceleration due to gravity = 32·2.

A more convenient form for hydraulic work is as follows :—

Area of orifice in square inches = $0{\cdot}0315\, Q/Cd \sqrt{P}$ where

Q is the quantity in gallons per minute.
P is the pressure in lb./sq. in.
Cd is the coefficient of discharge as before.

Diameter of a mushroom valve having a lift of 1/20 of the diameter, when the required area is known.

$$D = 2{\cdot}5 \sqrt{A} \text{ (approx.)}, \quad \text{where}$$

D is the diameter of the valve in inches.
A is the area of the opening at full lift.

Force acting on a hydraulic ram = $\pi r^2 P$ pounds, where

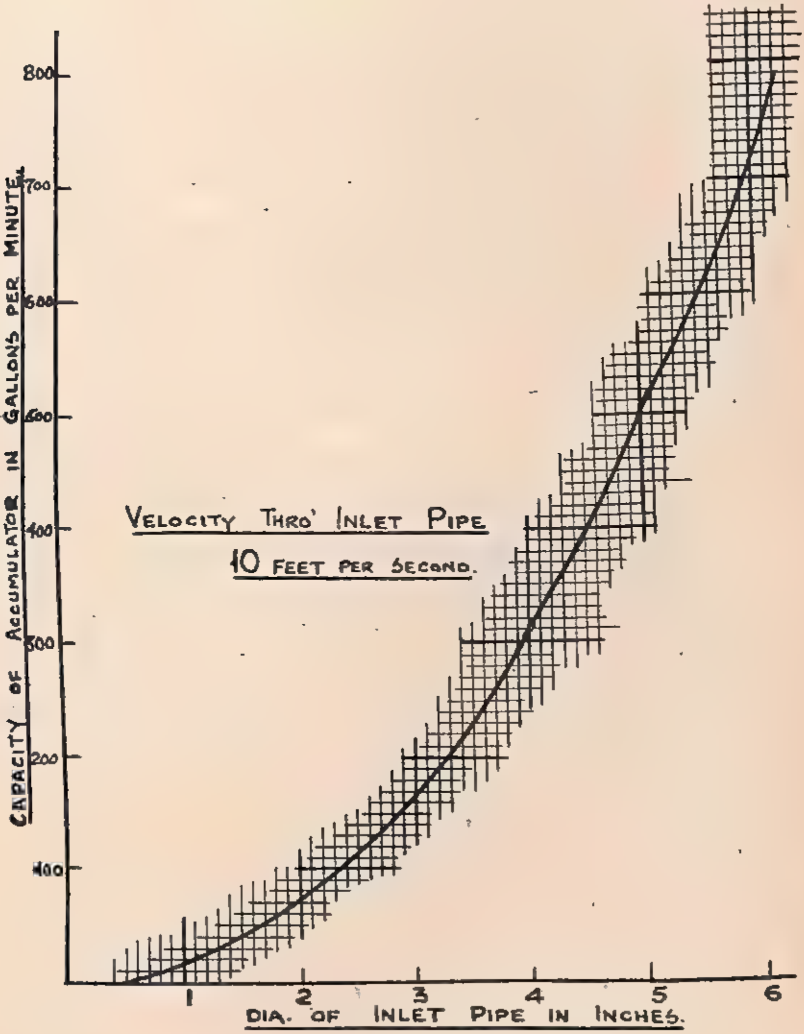
P is the pressure in lb./sq. in.,
r is the radius of the ram.
π is 3·142 (approx.).

APPENDIX B.

DIAMETERS, AREAS, AND EQUIVALENT IMPERIAL GALLONS PER FOOT LENGTH.

Dia. (Ins.)	Area (Sq. Ins.)	Galls. per Foot.	Dia. (Ins.)	Area (Sq. Ins.)	Galls. per Foot.	Dia. (Ins.)	Area (Sq. Ins.)	Galls. per Foot.	Dia. (Ins.)	Area (Sq. Ins.)	Galls. per Foot.	Dia. (Ins.)	Area (Sq. Ins.)	Galls. per Foot.
⅛	·0122	·0005	3¼	8·295	·3583	8¼	53·45	2·309	13¼	137·8	5·952	18¼	261·5	11·297
¼	·0490	·0021	3½	9·621	·4156	8½	56·74	2·451	13½	143·1	6·182	18½	268·8	11·612
⅜	·1104	·0047	3¾	11·04	·4769	8¾	60·13	2·597	13¾	148·4	6·410	18¾	276·1	11·927
½	·1963	·0084	4	12·56	·5426	9	63·61	2·747	14	153·9	6·649	19	283·5	12·247
⅝	·3068	·0132	4¼	14·18	·6125	9¼	67·20	2·903	14¼	159·4	6·886	19¼	291·0	12·571
¾	·4417	·0190	4½	15·90	·6868	9½	70·88	3·062	14½	165·1	7·132	19½	298·6	12·900
⅞	·6013	·0259	4¾	17·72	·7655	9¾	74·66	3·225	14¾	170·8	7·388	19¾	306·3	13·232
1	·7854	·0339	5	19·63	·8480	10	78·54	3·393	15	176·7	7·633	20	314·1	13·569
1⅛	·9940	·0429	5¼	21·54	·9348	10¼	82·51	3·564	15¼	182·6	7·888	20¼	322·0	13·912
1¼	1·227	·0530	5½	23·75	1·026	10½	86·59	3·740	15½	188·6	8·147	20½	330·0	14·256
1⅜	1·484	·0641	5¾	25·96	1·121	10¾	90·76	3·920	15¾	194·8	8·415	20¾	338·1	14·614
1½	1·767	·0763	6	28·27	1·221	11	95·03	4·105	16	201·0	8·683	21	346·3	14·960
1⅝	2·073	·0895	6¼	30·67	1·325	11¼	99·40	4·294	16¼	207·3	8·955	21¼	354·6	15·320
1¾	2·405	·1038	6½	33·18	1·433	11½	103·8	4·484	16½	213·8	9·236	21½	363·0	15·681
1⅞	2·761	·1192	6¾	35·78	1·545	11¾	108·4	4·682	16¾	220·3	9·516	21¾	371·5	16·050
2	3·141	·1356	7	38·48	1·662	12	113·0	4·881	17	226·9	9·802	22	380·1	16·420
2¼	3·976	·1717	7¼	41·28	1·783	12¼	117·8	5·088	17¼	233·7	10·095	22½	397·6	17·176
2½	4·908	·2120	7½	44·17	1·908	12½	122·7	5·300	17½	240·5	10·389	23	415·4	17·945
2¾	5·939	·2565	7¾	47·17	2·037	12¾	127·6	5·512	17¾	247·4	10·687	23½	433·7	18·735
3	7·068	·3053	8	50·26	2·171	13	132·7	5·732	18	354·4	10·990	24	452·3	19·539

APPENDIX C.



Inlet Pipe—Accumulator Capacity Ratio.

BIBLIOGRAPHY.

"Hydraulics and the Mechanics of Fluids." E. H. LEWITT. (Pitman).

"Recent Developments in High-Speed Reciprocating Pumps," by F. H. TOWLER and J. M. TOWLER. Paper to the Institution of Mechanical Engineers, December, 1937.

"Hydraulic Automotive Apparatus for Ships," by E. BRUCE BALL. Paper to the Institution of Engineers and Shipbuilders, January, 1942.

List of A.E.S.D. Printed Pamphlets and Other Publications in Stock.

*The Craft of Tracing, ...	
* (6d to members, others, 1/-. Post Free).	
Lost Volumes from Cones, ...	W. R. THOMSON
The Parallel Operation of Diesel-Alternator Sets, ...	G. H. BRADBURY
Introduction to Jig and Tool Draughting, ...	A. LEIGH
*Principles of Automatic Motor Control	B. FELDBAUER
* (1/6 to members, 3/- others. Post Free).	
Pumping and Flooding of Ships, ...	E. W. ANSELL
Power Station Auxiliaries, ...	T. H. CARR
Devices for the Prevention of Damage to Machinery,	A. CLARKE
Notes on Some Modern Methods of Structural Analysis,	F. SIMPSON
Static Rectification, ...	F. G. SPREADBURY
Magnetism in Industry and Science, ...	H. E. HUTTER
Modern Screw Propeller Geometry, ...	A. W. RIDDLE
Brakes for Locomotives and Other Rail Units, ...	GEORGE W. MCARD
Ward Leonard Control of Planer Type Machine Tools,	B. FELDBAUER
Surface Condenser Design, ...	ROBERT B. NEILSON
Reinforced Concrete, ...	J. MCHARDY YOUNG
Dynamic Balancing, ...	A. BELL
Fluid Film Lubricated Bearing Practice, ...	G. S. ELFER
*Power Station Civil Engineering and Building Works,	T. H. CARR
* (1/6 to members, 3/- others. Post free)	
The Current Rating of Paper Insulated Cables, ...	C. C. BARNES
Treatise on Ventilation and the Design of Ductwork for Air Flow, ...	THOS. W. BEARD
The Design of Structural Welded Details, ...	FRANK H. ABRAHAMS
The Recovery and Utilisation of Waste Heat, ...	J. A. HAGLEY
Machines for Dynamic Balancing, ...	RALPH M. EGGLESTON
An Introduction to the Construction of Nomographs,	WILLIAM W. CALDER
Foundations, ...	J. MCHARDY YOUNG
An Introduction to the Cathode Ray Oscillograph, ...	F. G. SPREADBURY
Panels and Panelling, ...	K E. HARPER
Design of Concrete Sections, ...	W. SCOTT WILSON
Industrial Ventilation and Air Treatment, ...	M. R. MORTON
Some Notes on the Stresses in Flat Rectangular Plates Under Pressure and The Design of Hopper and Suspended Type Bunkers, ...	GEORGE D. MOUAT
Electricity Sub-Stations, ...	T. H. CARR
The Analysis of Braced Frames by the Method of Shear Coefficients, ...	F. H. ABRAHAMS
Concrete Slab and Girder Floors, ...	W. SCOTT WILSON
Helical Gears for Marine Turbines, ...	DUNCAN CARMICHAEL
An Elementary Introduction to Industrial Electronics,	KENNETH J. FIELD
*Design of Transformers, ...	T. WATERHOUSE
*(1/6 to members, 3/- others. Post Free).	
Production of Accurate Screw Threads, ...	J. W. K. MURCH
High Pressure Hydraulic Control Equipment,	J. RODGER & R. HARTKOPF

All not otherwise marked are 1/- to members, 2/- others, post free.

List of A.E.S.D. Data Sheets.

1. Safe Load on Machine-Cut Spur Gears.
2. Deflection of Shafts and Beams
3. Deflection of Shafts and Beams (Instruction Sheet) } Connected.
4. Steam Radiation Heating Chart.
5. Horse-Power of Leather Belts, etc.
6. Automobile Brakes (Axle Brakes)
7. Automobile Brakes (Transmission Brakes) } Connected.
8. Capacities of Bucket Elevators.
9. Valley Angle Chart for Hoppers and Chutes.
10. Shafts up to 5½-in. diameter, subjected to Twisting and Combined Bending and Twisting.
11. Shafts, 5¾ to 26-inch diameter, subjected to Twisting and Combined Bending and Twisting.
12. Ship Derrick Booms.
13. Spiral Springs (Diameter of Rd. or Sq. Wire).
14. Spiral Springs (Compression).
15. Automobile Clutches (Cone Clutches).
16. „ „ (Plate Clutches).
17. Coil Friction for Belts, etc.
18. Internal Expanding Brakes. Self-Balancing Brake Shoes (Force Diagram)
19. Internal Expanding Brakes. Angular Proportions for Self-Balancing } Connected.
20. Referred Mean Pressure Cut-Off, etc.
21. Particulars for Balata Belt Drives.
22. ⅞″ Square Duralumin Tubes as Struts.
23. 1″ „ „ „
24. ¾″ Sq. Steel Tubes as Struts 30 ton yield).
25. ⅞″ „ „ „ (30 ton yield).
26. 1″ „ „ „ (30 ton yield).
27. ¾″ „ „ „ (40 ton yield).
28. ⅞″ „ „ „ (40 ton yield).
29. 1″ „ „ „ (40 ton yield).
30. Moments of Inertia of Built-up Sections (Tables)
31. Moments of Inertia of Built-up Sections (Instructions and Examples) } Connected.
32. Reinforced Concrete Slabs (Line Chart)
33. Reinforced Concrete Slabs (Instructions and Examples) } Connected.
34. Capacity and Speed Chart for Troughed Band Conveyors.
35. Screw Propeller Design (Sheet 1, Diameter Chart)
36. „ „ „ (Sheet 2, Pitch Chart)
37. „ „ „ (Sheet 3, Notes and Examples) } Connected.
38. Open Coil Conical Springs.
39. Close Coil Conical Springs.
40. Trajectory Described by Belt Conveyors
41. Metric Equivalents.
42. Useful Conversion Factors.
43. Torsion of Non-Circular Shafts.
44. Railway Vehicles on Curves.
45. Chart of R.S. Angle Purlins.
46. Coned Plate Development.
47. Solution of Triangles (Sheet 1, Right Angles).
48. Solution of Triangles (Sheet 2, Oblique Angles).
49. Relation between Length, Linear Movement and Angular Movement of Lever. (Diagram and Notes).
50. „ „ „ „ „ „ (Chart).

51. Helix Angle and Efficiency of Screws and Worms.
52. Approximate Radius of Gyration of Various Sections.
53. Helical Spring Graphs (Round Wire)
54. ,, ,, ,, (Round Wire) Connected.
55. ,, ,, ,, (Square Wire)
56. Relative Values of Welds to Rivets.
57. Ration of Length/Depth of Girders for Stiffness.
58. Graphs for Strength of Rectangular Flat Plates of Uniform Thickness.
59. Graphs for Deflections of Rectangular Flat Plates of Uniform Thickness.
60. Moment of Resistance of Reinforced Concrete Beams.
61. Deflection of Leaf Spring.
62. Strength of Leaf Spring.
63. Chart showing Relationship of Various Hardness Tests.
64. Shaft Horse Power and Proportions of Worm Gears.
65. Ring with Uniform Internal Load (Tangential Strain) Connected.
66. Ring with Uniform Internal Load (Tangential Stress)
67. Hub Pressed on to Steel Shaft. (Maximum Tangential Stress at Bore of Hub).
68. Hub Pressed on to Steel Shaft. (Radial Gripping Pressure between Hub and Shaft).
69. Rotating Disc (Steel) Tangential Strain Connected.
70. ,, ,, ,, ,, Stress
71. Ring with Uniform External Load, Tangential Strain Connected.
72. ,, ,, ,, ,, ,, Stress
73. Viscosity Temperature Chart for Converting Commercial to Absolute Viscosities. Connected.
74. Journal Friction on Bearings.
75. Ring Oil Bearings.
76. Shearing and Bearing Values for High Tensile Structural Steel Shop Rivets, in accordance with B.S.S. No. 548/1934. Connected.
77. Permissible Compressive Stresses for High Tensile Structural Steel, manufactured in accordance with B.S.S. 548/1934.
78. Velocity of Flow in Pipes for a Given Delivery. Connected.
79. Delivery of Water in Pipes for a Given Head.
80. Working Loads in Mild Steel Pillar Shafts.
81. Involute Toothed Gearing Chart.
82. Steam Pipe Design. Chart showing Flow of Steam through Pipes.
83. Variation of Suction Lift and Temperature for Centrifugal Pumps.
84. Nomograph for Uniformily Distributed Loads on British Standard Beams.
85. ,, ,, ,, ,, ,,
86. ,, ,, ,, ,, ,, Connected.
87. Notes on Beam Design and on Use of Data Sheets Nos. 84-5-6.
88. ,, ,, ,, ,, ,,
89. Curve Relating Natural Frequency and Deflection.
90. Vibration Transmissibility Curve for Elastic Suspension. Connected.
91. Instructions and Examples in the Use of Data Sheets No. 89 and 90.
92. Pressure on Sides of Bunker.

93-4-5-6-7. Rolled Steel Sections.

98-99-100. Boiler Safety Valves.

(Data Sheets are 2d to Members, 4d to others, post free).

Orders for Pamphlets and Data Sheets to be sent to the Editor, *The Draughtsman*, cheques and orders being crossed "Draughtsman Publishing Co., Ltd."